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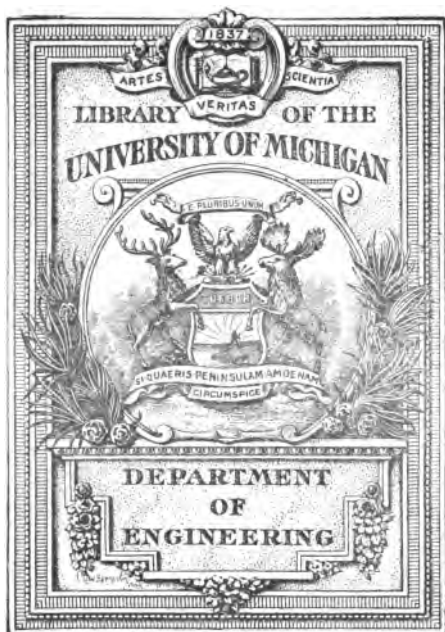
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A PRACTICAL TREATISE
ON THE
STEAM ENGINE INDICATOR
AND
INDICATOR DIAGRAMS:

WITH NOTES ON
STEAM ENGINE PERFORMANCES, THE EXPANSION OF STEAM,
BEHAVIOUR OF STEAM IN STEAM ENGINE CYLINDERS,
AND ON GAS ENGINE DIAGRAM.

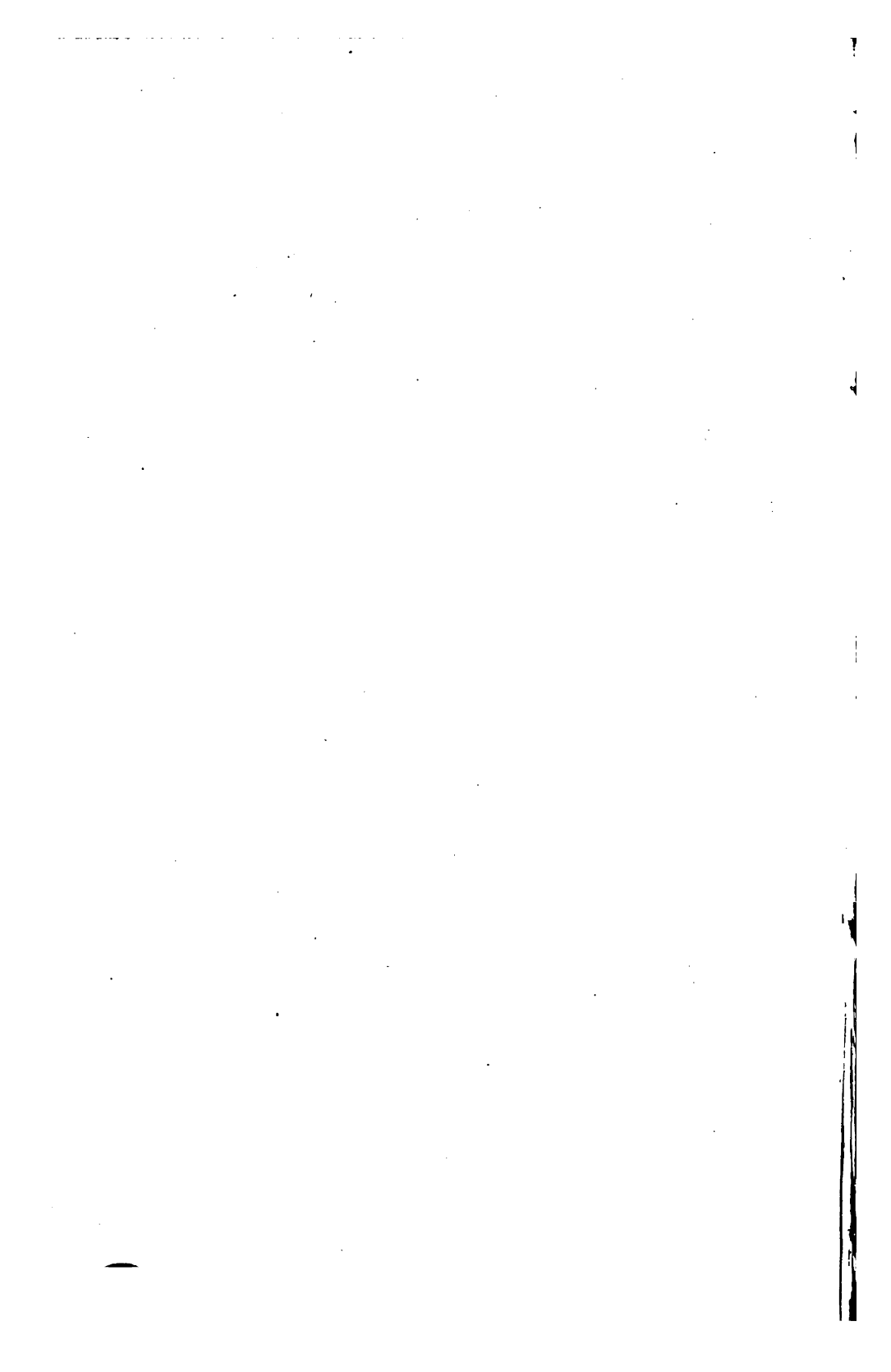
BY
A M I C E, *pseud.*

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PREFACE.

THE intention of the writer of this book has been to place in the hands of students and practical men a concise guide to the objects, construction and use of the Indicator, and to the interpretation of Indicator Diagrams. Care has been used to avoid lengthy discussion of theoretical or hypothetical matters which, although of great scientific importance, do not immediately concern those whose desire is to understand the practical employment of the Indicator as used by those interested in any way with the working of steam and other motors.

The plan upon which the Author has proceeded has been, firstly, to consider the object of an Indicator Diagram, or what it is desired that the diagram shall show; secondly, to describe the construction of the Indicator in its various forms, as successively devised for this purpose; thirdly, to describe the apparatus necessary for the attachment of the Indicator to the engine, and how to use the instrument; fourthly, to give examples of diagrams from all kinds of engines most in use, to interpret

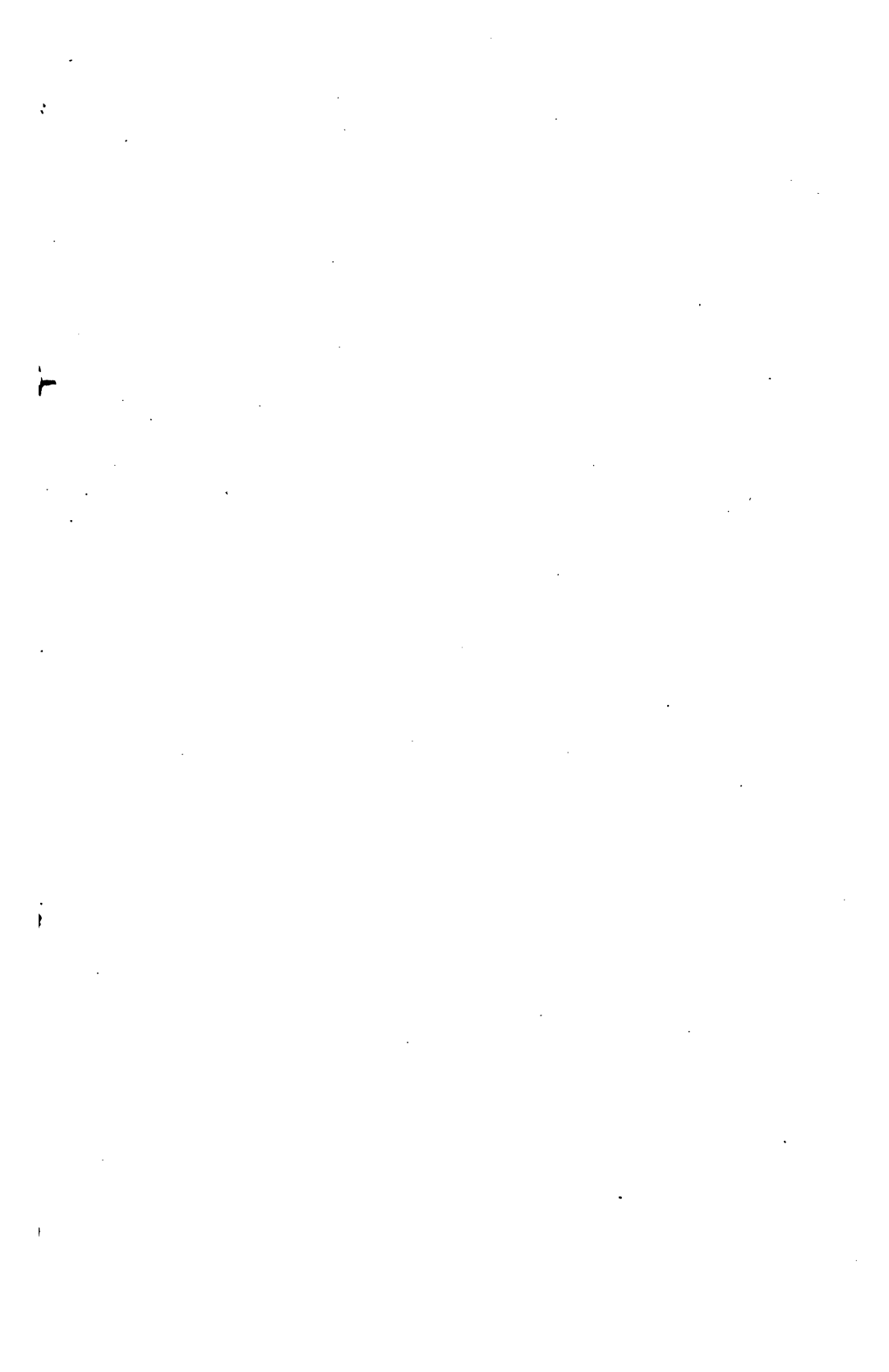
these diagrams, and to show how far they agree with theoretical diagrams ; fifthly, to show the most simple methods of calculating and constructing theoretical curves of expansion, and of comparing, in particular cases, the actual with the theoretical performance of steam in the steam engine cylinder.

In almost all cases simple arithmetical treatment has been adopted, so that the book shall be useful to practical men.

The thermo-dynamics of the steam engine or of other heat engines have not been considered as part of the subject of the book, but the behaviour of steam and its expansion under different conditions have been treated in a simple manner as far as these questions are important to the consideration of Indicator Diagrams in their most usual practical applications.

Some chapters of the book have appeared in the pages of *The Electrician*, but the greater part appears here for the first time.

Lastly, it has not been considered necessary to add to the book by again publishing tables of areas of circles or of steam pressures and temperatures, tables which are to be found in so many books, some of which are in the hands of every reader likely to require this work.



THE STEAM ENGINE INDICATOR.

PART I.

OBJECTS OF THE INDICATOR DIAGRAM AND CONSTRUCTION OF INDICATORS.

The indicator may be looked upon as a mirror which shows us what is going on in the cylinder of a steam engine, both qualitatively and quantitatively. It is of course equally applicable for the same purpose to any other form of gas or fluid pressure motor. We shall, however, except where specially stated, speak generally of the steam engine.

Like almost all the essentials of the steam engine, the indicator was invented by Watt. He designed it for ascertaining the work done by the steam on a steam engine piston, and although in the hands of modern investigators, and with the aid of thermo-dynamics, it has performed most important services other than that of determining the dynamic value of the work done by steam, it is still most largely used for that more simple purpose.

The work done by a steam engine depends on the velocity of its piston and the pressure by virtue of which it acquires and maintains that velocity, these elements being usually expressed in feet per minute and pounds per square inch. That is to say, in foot pounds per minute. Of these elements the indicator diagram shows primarily one—namely, the pressure, and its function is especially to show the variations in this pressure due to expansion and to other causes of change of pressure during the stroke of the piston, and therefore to show variation of pressure through a distance. If the pressure in the cylinder were uniform throughout the stroke, the indicator diagram would not be required as an exponent of pressure,

but it would still be required in some part of the history of every engine, for the purpose of showing whether in any or in what parts of the stroke variation occurred, if at all.

It is necessary here to say a few words on what is meant by work, and how this can be graphically represented. There must be a clear understanding of the connection between the indicator diagram and the indicated work. It must be seen that the indicator diagram only represents graphically the pressure acting on the piston of a steam engine during one stroke. It thus represents pressure, and distance throughout which the pressure acts, and therefore work done; but it does not show at what rate that work was done, and, therefore, not horse-power, inasmuch as the latter is expression for rate of doing work. That is to say, one term in the element, velocity, namely, time, has to be supplied. The diagram shows that pressure acted on the piston throughout the length of a stroke, but whether a century of years or only a second of time were occupied in performing this stroke it does not show. In either case the amount of work done would be the same,* but if it were only accomplished in the century, it would be useless for practical purposes. Thus, to use Professor Tait's definition, "the horse-power of an agent, or the amount of work done by an agent in each second, is the product of the force into the average velocity of the agent." If an agent, as an engine, raises 1,000lbs. through 10ft., it does $1,000 \times 10$ or 10,000 foot pounds of work, but the power of that engine will depend upon the time it takes to do that work, or its rate of doing work, and the horse-power is the power of doing 33,000 foot pounds of work in a minute, or 550 foot pounds per second.

Of the power of the animal whose capacity for work provides this unit, everyone has a natural conception. The horse is so

* This is not absolutely accurate, but for the velocities occurring in steam engine practice it is not necessary to consider the difference between the work done by the agent which, for instance, raises 1,000lbs. one foot in a second and that which raises 500lbs. in half the time through the same height.

universally employed for doing work that every man acquires insensibly a notion of the amount of work one or a number will do. Previous to the introduction of the steam engine horses were used to do the work subsequently done by that agent. Watt therefore set to work to find a numerical value for the work done by a good horse in a unit of time, and found it to equal 33,000lbs. raised 1 foot in 1 minute.

A graphic representation of the work done by steam on the piston of a steam engine would, if the steam were admitted at full pressure throughout the stroke of that piston, be afforded by a simple rectangular parallelogram. It would be thus

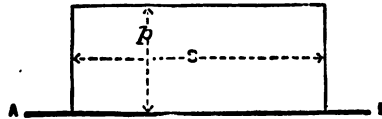


FIG. 1.

in the case of a high pressure non-condensing engine ; while in a condensing engine of similar length of stroke and using the same pressure of steam it would be thus

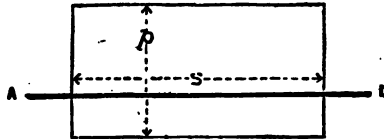


FIG. 2.

In both these diagrams S represents the stroke of the piston and p the pressure acting upon it. The line $A-L$ represents what is, with the high pressure non-condensing engine, the line of no pressure, or what in reality represents the pressure of the atmosphere. The whole of the diagram is formed by steam at a pressure above that of the atmosphere. In the condensing engine it also represents the atmospheric pressure, that part of the diagram which is below that line representing the proportion of the pressure of the atmosphere which is made available as pressure acting on the piston by the formation of

a vacuum in the condenser, this pressure being added to the steam pressure acting on the other side of the piston. The popular conception of the action of a vacuum is that of a power of suction which pulls the piston, but its action is due to the removal of atmospheric pressure from one side of the piston, so that the result is equivalent to making the pressure of the atmosphere available for doing work on the other side of the piston.

Before proceeding to the indicator we may define what we want it to show. Assume now that the line $A-L$ in Fig. 3 is drawn on a piece of paper at such a level that it would as concerns non-condensing engines represent the line of no pressure. Although, as above explained, it would represent the pressure of the atmosphere, or say 15lbs. per square inch, this pressure is not available by the non-condensing engine, and is therefore to it a zero pressure, just as it is with an ordinary steam boiler, the zero point on the steam pressure gauge dial being the point at which the pointer stands when the gauge is subject to the atmospheric pressure. Turning

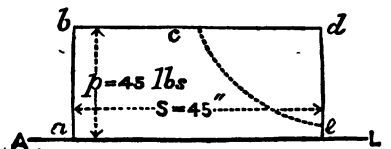


FIG. 3.

to Fig. 3, then, we will assume that the steam pressure from the boiler is such that it gives a pressure, p , of 45lbs. per square inch in the cylinder. Steam being admitted to the cylinder at the commencement of the stroke S of the piston, we may draw a line, ab , normal to AL , and of such a length that with a convenient scale it represents 45lbs. This, then, is the pressure per square inch acting on the piston at the commencement of the stroke. Assuming that the full pressure steam is admitted throughout the stroke, a line c , parallel with AL , and of a length $bd = S$, representing the whole stroke of

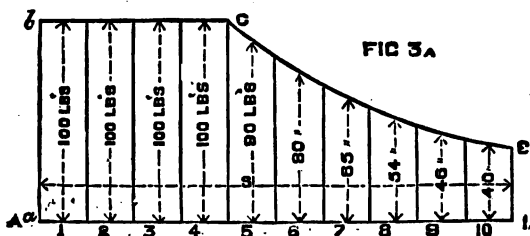
the piston, will show the pressure at every part of the stroke made in the direction bd (in our case assumed uniform). The piston having reached the end, d , of its stroke, and exhaust being supposed to take place instantaneously, the line de may be drawn, indicating that the pressure at the end of the stroke has fallen to atmospheric pressure, or in this case to zero. The piston now commences the return stroke, and the pressure during this may be represented by the line ea . A parallelogram is thus completed which, if from an actual engine, would indicate that there was a uniform pressure throughout the stroke, and the indicated horse-power of an engine with a stroke, S , of $45'' = 3.75\text{ft.}$ and cylinder diameter, $D = 20.5\text{in.}$ in diameter, and having an area, $A = 330\text{in.}^2$,* the engine making 50 revolutions, R , per minute, and being double-acting—that is, having steam pressure alternately on each side of the piston—would be

$$\frac{D^2 \pi P (S \times 2) R}{4 \times 33,000} = \frac{A P R (S \times 2)}{33,000} = \frac{330 \times 45 \times 7.50 \times 50}{33,000} = 168.7 \text{ h.p.}$$

It must here be stated that the full pressure of steam is never admitted throughout the whole stroke, but is cut off, according to circumstances and the kind of work done by the engine, at an earlier part of the stroke, as, for instance, one-half, when the pressure falls as shown by the curved dotted line ce , the piston being moved and the work performed from the point of cut off—namely half stroke—by the expansive energy of the steam, admitted previous to that point. With expansion through given ranges we shall deal further on. The diagram obtained under these circumstances is therefore not a parallelogram. The pressure acting on the piston is not constant. The vertical height of the line ab varies at different parts of the distance S , and hence to find the work done during the passage of the piston through that distance, we

* Usually the sectional area of the piston rod should be deducted from this. If this piston rod only passes through one cylinder cover, as is the more common, then the deduction from the area of the piston should only be one-half the sectional area of the rod, as it only reduces the effective area of the piston on one side.

must divide S into a number of equal intervals, and, assuming the pressure uniform throughout each of these, construct a parallelogram within each interval and the bounding pressure line b, c, e . The work done is then represented by the combined area of the several small parallelograms. Thus, suppose S (in Fig. 3A) is divided into 10 equal parts by 9



intermediate lines. During the first four of these the height $a b$ represents, say, 100lbs.; the remaining six have mean heights which successively represent, say, 90lbs., 80lbs., 65lbs., 54lbs., 46lbs., and 40lbs. Now if $S=10$ ft., each interval being 1ft., we shall have for work done Mean pressure \times 10ft.

$$= \frac{(100 \times 4) + 90 + 80 + 65 + 54 + 46 + 40 \text{ lb}}{10} \times 10 = 77.5 \text{ lb} \times 10 \text{ ft}$$

or the diagram has an area of 775, showing a mean pressure throughout the stroke of $\frac{775 \text{ lbs.}}{10} = 77.5 \text{ lbs.}$, and this multiplied

by the stroke, which, in our case is 10ft., represents 775 foot pounds. From numerous causes the diagram actually obtained is never square at the corners d , Fig. 3, or at e , Fig. 3A, and not often at a , but into these causes we shall enter at the proper time.

Watt's indicator should be here described, as it may be taken as illustrating the simplest form of the instrument, and our younger readers will be assisted by reference to its construction. Several of these indicators are still in existence, and the accompanying engravings from the *Proceedings* of the

Institute of Mechanical Engineers, 1883, is made from one of these.

Although, generally speaking, and as concerns practical work, the indicator as made by Watt is of little more than historic interest, it is of considerable educational value, because a knowledge of this, as the simplest form of indicator, will make the principle of the instrument more easy of acquirement than if described by reference to its modern representative, while an explanation of its defects will make clear the reasons for the steps in the development of those now most used. It will therefore be described here at length, not because of its historic interest, but as being necessary to our purpose.

From our engravings, Figs. 4 and 5, it will be seen that a cylinder, A, is fitted with a piston, B. To this is attached a piston rod, P, which at its upper end carries a pencil, D. The piston B moves upward in the cylinder, under the action of steam admitted by the cock H. Above the piston is a spring, C, the strength of which is known; that is to say, the number of pounds which is necessary to compress it through a given range is ascertained before it is used in the indicator. Suppose, for instance, it takes a weight of 10lb. to compress the spring one inch; then when it is placed in the indicator it will require 10lb. to do the same compression, and if the cylinder be one inch in diameter, and consequently 0.7854 " sectional area, it will require a steam pressure of

$\frac{10}{0.7854}$, or 12.719lb. per square inch to compress the spring and raise the piston one inch. In practice, however, the springs are made of such strength that they will compress through a range of one inch with a pressure represented by a convenient number, and in the above case the spring would be of such strength that with a steam pressure of 10lb. per square inch it would compress one inch, and therefore with a load of $10 \times 0.7854 = 7.854$ lb.

Mounted with slides in the frame F, as seen in Fig. 5, is a board, E, to which is attached the piece G. Fastened to G is one piece of string, S, running over a small pulley, and carrying

the weight *W*, and another piece of string, *S'*, running over

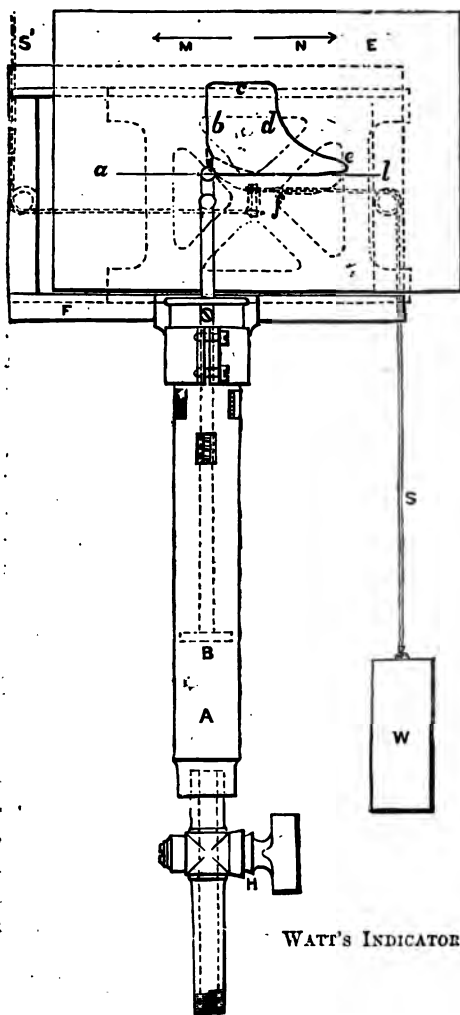


FIG. 4.

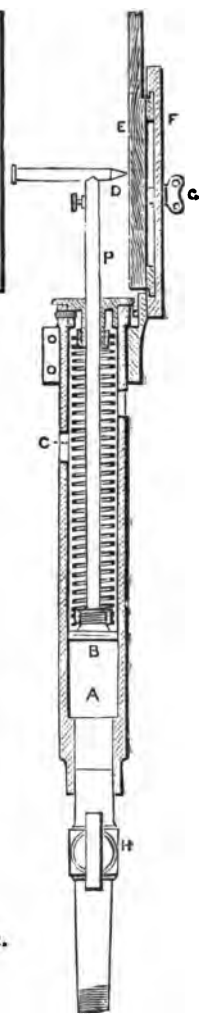


FIG. 5.

WATT'S INDICATOR.

another pulley and attached to some part of the engine which

has a reciprocating movement synchronous with that of the piston of the engine. The board E is thus pulled in the direction M by the string attached to some part of the engine, and as that part returns with the return stroke of the engine piston, the weight W pulls the board back in the direction N. The weight W thus acts in pulling the board E in one direction, and also prevents any slack in its movement. The cock H being screwed into one end of the steam engine cylinder, the cylinder A of the indicator receives steam at the same time as the engine cylinder, and at the same time the board E is put into motion by the engine. A piece of paper is attached to the board E, and the board allowed to be moved by the engine before steam is admitted to the indicator cylinder A. The pencil being lightly pressed against the board, the line *a l* will be made, and will represent the pressure of the atmosphere. It will now be seen that the engine piston being at the end of its stroke, and steam being admitted to the cylinder, it will also be admitted to A, and the pencil D will instantly rise and trace the line *b*, which will have a height according to the steam pressure. The engine piston being now in motion, the board E is pulled in the direction M, and the horizontal line *c* is traced. This line will be at a height above the line *a l*, which will represent the maximum pressure per square inch in the steam engine cylinder, and if the ports in that cylinder are of sufficient size and the movement of the slide valve sufficiently rapid, this pressure will be maintained in the cylinder until near the point at which the entrance of steam into the cylinder is cut off by the slide valve. From that point the pressure of the steam falls in the cylinder, as shown by the curved line *d*, this line being the expansion curve, or the line which shows the pressure existing in the cylinder at any part of the stroke of the piston during the expansion of the volume of steam admitted up to the point of cut off. At *e* the exhaust port is opened, and steam instantly rushes into the atmosphere, or into the condenser, if the engine is fitted with the latter, and the pressure falls correspondingly, and the pencil with it. At the same time the return stroke of

the engine commences, and the weight *W* pulls the board *E* back as that stroke is made, and the pencil *D* traces the line, completing the diagram, this line being nearly parallel with and nearly coincident with the atmospheric line *a l*, if the engine is non-condensing. If the engine is fitted with a condenser the pressure falls below the atmospheric pressure; in other words, a vacuum more or less complete is made by the condensation of the steam, and the line *f* is traced. In this case the pressure against the engine piston is represented by the distance between the line *f* and *c d*, instead of between the line *a l* and *c d*. To the form in detail of the diagram we shall recur hereafter.

The steam engines made by Watt were all very slow speed engines, and a low pressure of steam was used. Under these conditions a spring of rather long range and with the pencil attached direct could be used. The return motion of the indicator paper or card could also be effected by a weight, as shown. With the use of high pressures and high speeds, however, it became necessary to alter these parts. In the first place, board *E* of the Watt indicator was large and inconvenient, and the weighted string acted in a jerky manner, making the movement of the board unsteady, especially in the direction *N*. To overcome these objections, the indicator known as M'Naught's was made. In this the range of movement of the pencil was, like Watt's, the same as that of the compression of the spring, but the board *E* was replaced by a vertical hollow cylinder turning upon a vertical axis. This cylinder contains a spring, which takes the place of the weight *W* in Watt's indicator, and the paper or card being placed upon it, the pencil traces the diagram exactly as on the flat board. This improvement did not remove the defect due to the use of a long range spring and of a pencil attached thereto, the effect of which was that with high pressure steam and quick running engines the instantaneous compression of the spring was followed by a series of elastic oscillations, sometimes of considerable range, aggravated by the inertia of motion of the piston and attached parts, and

causing the pencil to trace an irregular line, which reduced the value of the diagram as an indication of the behaviour of the steam in the cylinder. The pencil would trace a line as shown at *b* (Fig. 6), making it difficult to obtain the mean pressure line, which would probably be as indicated by the dotted line. In very high speed engines the whole of the line from *b* to *c* (Fig. 6) was of an irregular character, giving evi-



FIG. 6.

dence of tremulous movements due to the indicator itself, and not at all due to the engine. To remedy these defects Mr. C. B. Richards, an engineer living at Hartford, Connecticut, designed the indicator known by his name. McNaught's indicator is only to a small extent employed now, other forms are much more used, the Richards indicator being still the most widely known.

We have given the reasons which make it necessary to employ, in an indicator for taking diagrams from modern engines, a spring of short range and a piston as light as possible, and light attached parts. These requirements were, as far as seemed necessary at the time, embodied in Richards's indicator, of which we now give an engraving (Fig. 7), showing the indicator as originally made. This gives but an external view, which is sufficient for our present purpose. It will be seen that the piston rod is attached by a short link to an arm forming part of a small parallel motion. This attachment is made at one-fourth of the length of the arm from its fulcrum on the indicator frame. The end of this arm has, therefore, a range of movement four times that of the indicator piston,

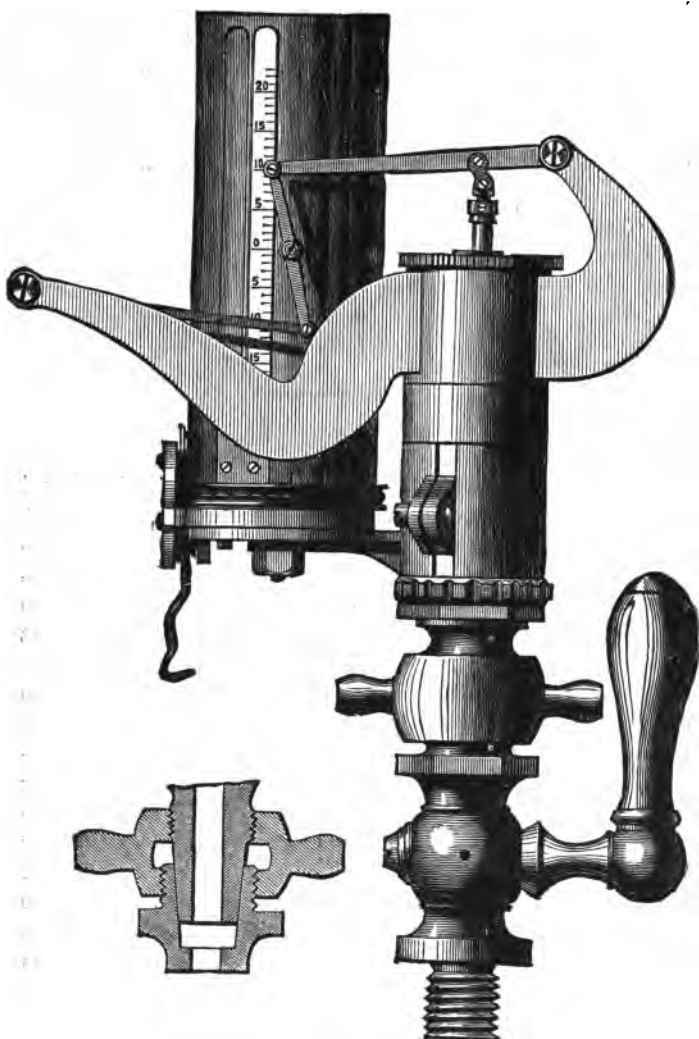
The Richards Indicator, 1868.

FIG. 8

FIG. 7.

and, consequently, the pencil which is carried at the centre of the middle member of the parallel motion has the same range. The indicator card, or paper on which the indication is taken, is placed upon the drum seen behind the pencil, and held by the light spring clip; one part of which is shown graduated. The strong curved arm carrying the parallel motion is free to turn upon the outer part of the indicator cylinder, and thus the pencil may be withdrawn from or put into contact with the drum on which the paper is wound. By means of the cord seen below the paper drum, and in the grooved sheave at the bottom of the drum the latter is attached to some part of the engine, so as to receive motion synchronous with the steam engine piston. The return motion of the drum is effected by a coiled spring within the drum. The string is thus always kept tight under the tension due to the spring in the drum. As the velocity of recoil of a steel spring is about 1,300 feet per second, no engine piston moves fast enough to cause the slack in the string which produced the jerky motion in the return stroke as effected by a weight in the Watt indicator. The small engraving (Fig. 8) is a section of the upper part of the cock which is screwed into the steam-engine cylinder or a pipe attached thereto, and of the lower part of the indicator cylinder with the fly-nut by which the indicator is attached to the cock. Two of these cocks may be employed, one fixed on either end of the engine cylinder. To take a diagram from either end of the cylinder it is, then, only necessary to remove the indicator from either fixed cock by unscrewing the fly-nut, the taper end of the indicator cylinder being steam-tight in the upper part of either cock. Instead of this it is common when a number of diagrams have to be taken to connect the two ends of the cylinder by a pipe with one three-way cock in the centre, to which the indicator points; it is then only necessary to turn the cock one way or the other to take a diagram from either end.

This indicator was a very great improvement on the best that had preceded it, and is at the present time very largely in use with but minor improvements. These improvements

are, however, of considerable value, and as it is not necessary to show them separately, we may show them by reference to the recent form of Richards indicator shown in Fig. 8A and

THE RICHARDS INDICATOR, 1882.

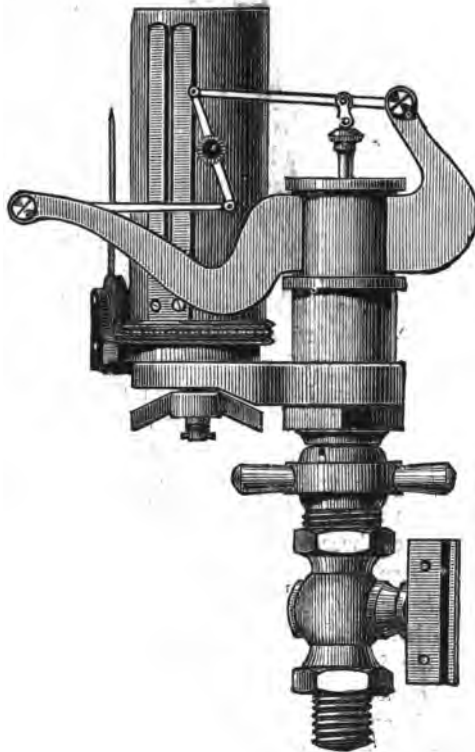


FIG. 8A.

to a modification of the Richards indicator by Mr. Richardson, by means of which a continuous series of indications may be taken on a long strip of paper. This indicator is an improvement upon one made by MM. Guinotte and De Hennault, also introduced by Messrs. Elliott Brothers, and is shown by Fig. 9, and

Fig. 10 is illustrative on a small scale of a series of diagrams taken from an engine with Corliss's valve gear. One of the improvements consists in the use of the fly-nut below the paper drum by which the part carrying the little pulleys

RICHARDSON'S CONTINUOUS INDICATOR.



FIG. 9.

which guide the string can be without loss of time adjusted to any position which may be necessary to place them in a line with the string when attached to the moving part of a steam engine. A second improvement consists in the use of a device for instantaneously stopping or

starting the paper drum without disconnecting the string by which the motion is communicated to the drum. With slow running engines the latter is not at all difficult, but with high speeds it is not only sometimes difficult to disconnect or connect the string to the indicator while the engine is running, but is sometimes dangerous. The device by which this is overcome is known as Darke's patent detent. It is simple, and consists of a ratchet segment on the bottom of the paper drum, and a small pawl controlled by a light spring attached to the outside of the cylinder. The pawl is seen in Fig. 9, upon a short pedestal between the drum and cylinder. As shown, it is out of gear, but if the string were pulled and the drum turned as in work, so as to bring the ratchet teeth round about three-fourths of a revolution from the position shown, the pawl would, by moving the little knob on the spring in front of the



FIG. 10.—Diagrams from Continuous Indicator.

cylinder, go into gear, and the drum would be held by it against the resistance of the drum spring. The string would then hang slack during the return stroke of the reciprocating part of the engine by which the string is moved. These two engravings show the distinguishing features of the Richards indicator as far as external parts are concerned; but, as already observed, it was not only the use of the parallel motion which formed a feature, but the use of a short spring, with light internal piston. To these springs we shall refer further on. For the present, having the continuous indicator before us, we may briefly describe it. It is in form a Richards indicator, but is somewhat larger, more especially in the drum. In the interior of the latter is a receptacle for a roll of paper, the end of which is brought through a slot. It is then passed round the cylinder, and is

inserted again into the interior, when it is caught by a slotted roller, which roller is worked in one direction during the motion of the paper drum. The backward and forward rotation of the drum gives intermittent motion to a train of small gearing, part of which is seen in the upper part of the drum. By means of the notched disc at the top of the drum, and the little spring catch lever which takes into the notches, diagrams may either be taken continuously, or if necessary single diagrams may be taken in the same way as an ordinary Richards indicator, or by throwing the drum out of gear by the Darke's detent they may be taken occasionally. When the notched disc is set fast by the spring catch shown in one of the notches, the small train of gearing by which the roll of paper is actuated is in gear, but when the catch is not in the notch the whole of the drum and its contents work just as an ordinary Richards indicator, and single diagrams can be taken in the ordinary way. When the diagrams have all been taken, the length of paper can readily be pulled off the roller. The paper is supplied in rolls, and the trouble of coiling it on the cylinder is entirely avoided. About 150 diagrams can be taken on each roll of paper.

Modern practice has continually increased not only the speed of engines but the pressure at which they work, and the necessity has arisen for an indicator which has moving parts for carrying the pencil even lighter than those in that of Richards. The reasons which made Richards' invention necessary have again made it necessary to improve the apparatus. Not only do the high speeds and pressures tend to break the parallel motion, but the several parts spring a little, and this with slight looseness in the joints, together with the inertia of the whole, conspires to increase the unsteadiness of the pressure line, giving to it a fluctuating waving form which is not a faithful representation of the real pressure in the cylinder. The total error, however, has probably been exaggerated, except with respect to what are now known as high speed engines, but a modification has become sufficiently imperative to induce Messrs. Elliot Brothers, of Charing-cross,

who have been the sole makers of the Richards indicator, to bring out the form of indicator known as Darke's patent; and

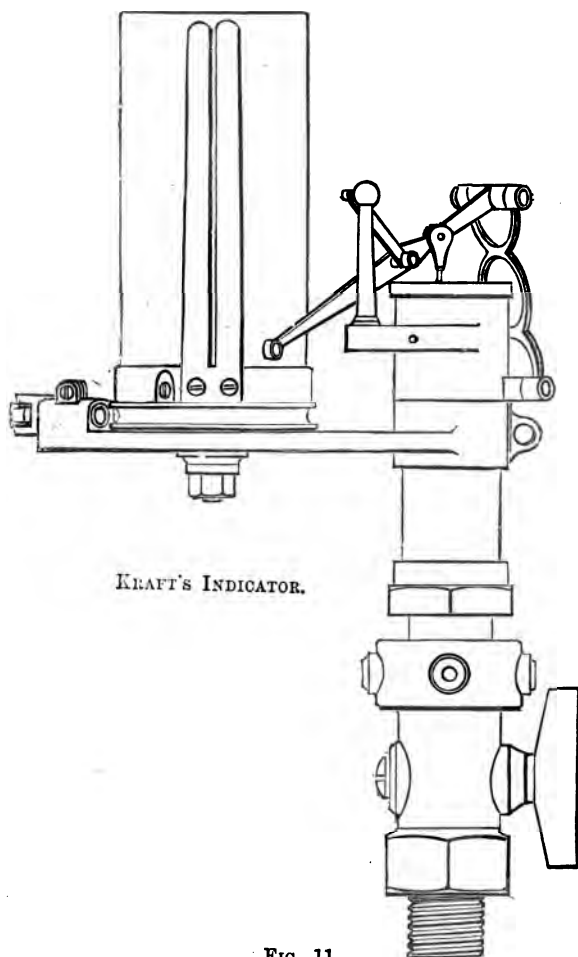
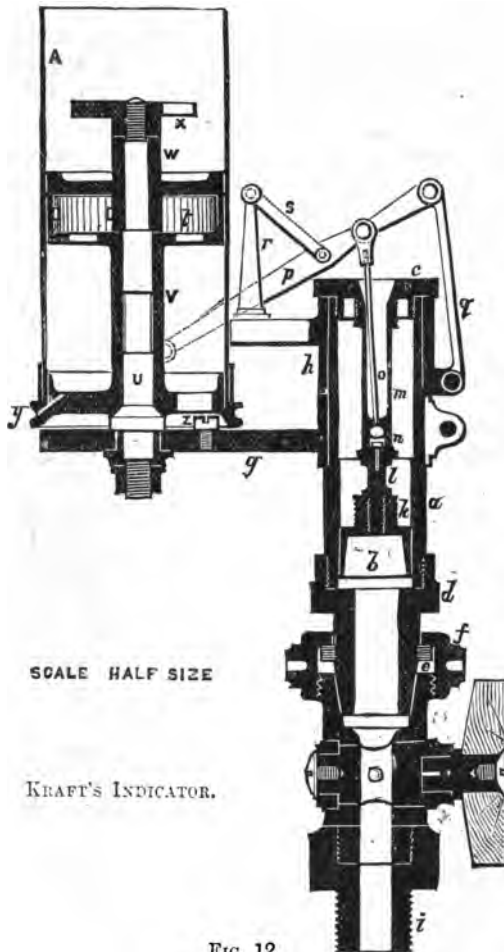


FIG. 11.

even the Richards indicators now made by them are fitted with Darke's pencil motion instead of the parallel motion, unless

otherwise ordered. We will therefore, before entering upon the use of the steam engine indicator, describe this latest



development of the instrument by these well-known makers ;
but inasmuch as an indicator, known as Kraft's, and having a
o 2

pencil arm with a free end and other points of interest, appeared before Darke's, we must first describe and illustrate this.

Kraft's indicator is illustrated by the accompanying perspective and sectional engravings (Figs. 11 and 12). This was one of the first indicators made, with a very light pencil arm having a free end. The other moving parts were also reduced to the smallest weight. In our engraving (Fig. 12), *a* is the steam cylinder, fitted with a piston, *b*, the piston being, as is usual in these instruments, made steam tight by a very accurate fit, the cylinder being very carefully bored. No rings or means of making the piston tight, except accuracy of workmanship, are ever used for indicator pistons. The upper part of the cylinder is slightly enlarged and closed by the cover *c*, while the lower part is firmly screwed into the tapered end piece *d*, which fits into the upper part of the indicator cock at *e*, where it is held by the hand nut *f*. The upper part of the cylinder is surrounded by a collar, *h*, which carries an ear by which the standard *r* is supported, and a projection to which *q* is articulated, the right line motion of the pencil being obtained by the radius rod and the proportion between its length, the length of *q* and the position of the joints in the pencil arm *p*. To prevent water from sprinkling the diagram paper, as it does when holes for the admission of air above the piston are made in the top cover, they are in this indicator made below the piece *g* which carries the paper drum *A*. The piston rod *l* is of a trunk form at *m*. The spring, not shown, against the resistance of which the steam raises the piston, is of spiral form, and is provided with a nut at either end, by which it is screwed to the piston at *k* and to the inwardly projecting boss in the under side of the cover *c*. The trunk *m*, or tubular part of the piston rod, slides through the cover *c*. Within the trunk is the rod *o* jointed to the pencil arm *p*, the lower end of this rod having a ball and socket joint. Our engraving shows the indicator at half size, so that it will be seen that the moving parts are all light. The pencil arm is generally made to carry a metal pencil—for instance, soft brass, to mark

on prepared diagram paper. It is, however, also made to take a very hard lead pencil, not only so that ordinary paper may be used, but because some German engineers consider that the friction of a hard lead pencil on glazed paper is less than that of a metal pencil on the prepared paper known as metallic paper, which is more generally used. There is no doubt that the friction of a metal pencil is greater on the prepared paper than a hard graphite pencil on hard paper, especially when the prepared paper, which is much the more expensive, is in a steamy atmosphere.

The paper drum A moves upon a steel spindle U, screwed into the projection *g* by an ordinary nut. The tube V is loose upon this spindle, and is made in one piece with the pulley *y* and the box *z*, which contains a coiled spring. The piece W is a cover for the spring, and is held in place by the nut X. The drum A is fixed to the pulley *y*, round which is coiled the string, by which motion is given to the drum from some part of the steam engine. The string is guided to the pulley *y* by a pulley which may be set in any desired direction. The string pulls the drum round through about three-fourths of a revolution against the resistance of the spring at *z*, which forces it back as the string returns.

The plug of the cock to which the indicator is attached is drilled with holes in two directions, so that, as shown, steam may pass from the engine cylinder to the indicator piston, or by turning the cock in one direction steam may be allowed to blow through to the air from the pipe below the indicator, so that any water in the pipe connections may be blown away. If water is in these connections, especially if they are of any length, it will give to the diagram an irregular form, not representative of any action of the steam in the engine cylinder.

The piston springs constitute one of the most important parts of an indicator, and with these generally we shall deal specially when we have given the reader some further particulars of the indicators most in favour. The next in order is the indicator invented by Mr. E. T. Darke, and which is illustrated by Figs. 13, 14, 15, 16, and 17. One of the most

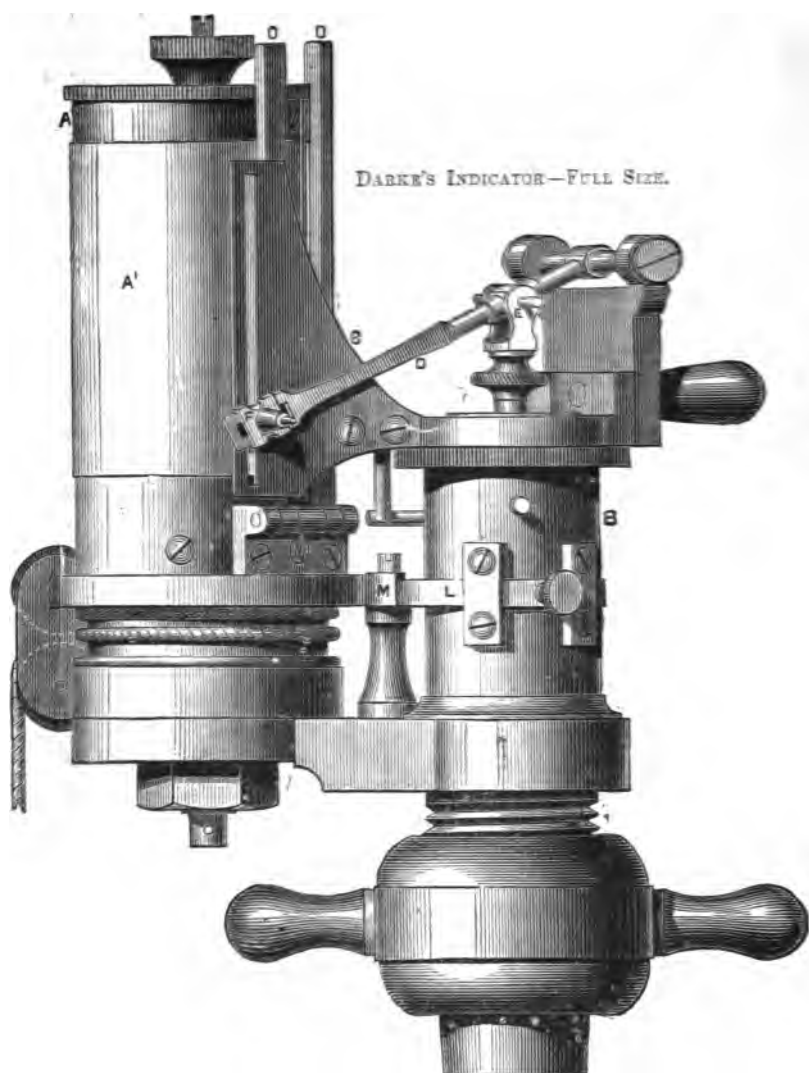


FIG. 13.

important features in this is the pencil motion, which consists of one steel arm pivotted at one end by centre points on the two ends of a T head, and carrying at the other a sliding piece, F, which is provided with a holder for a metallic pencil. The pencil is kept in a

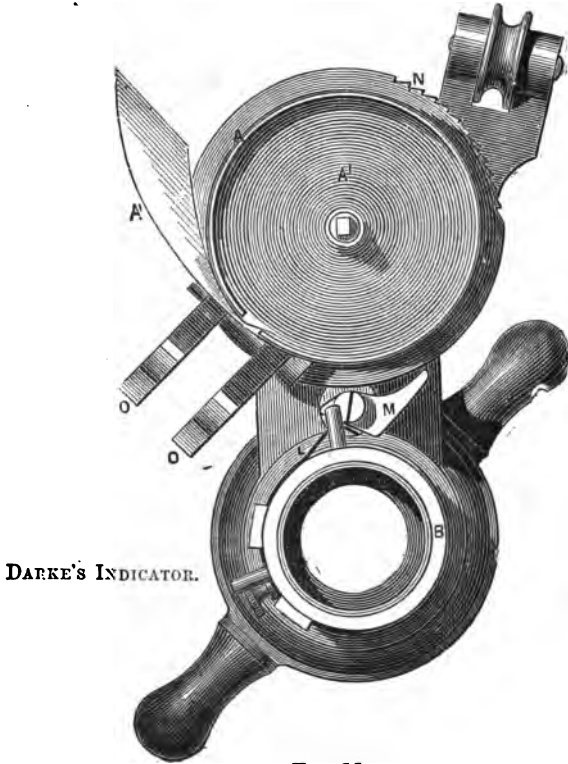
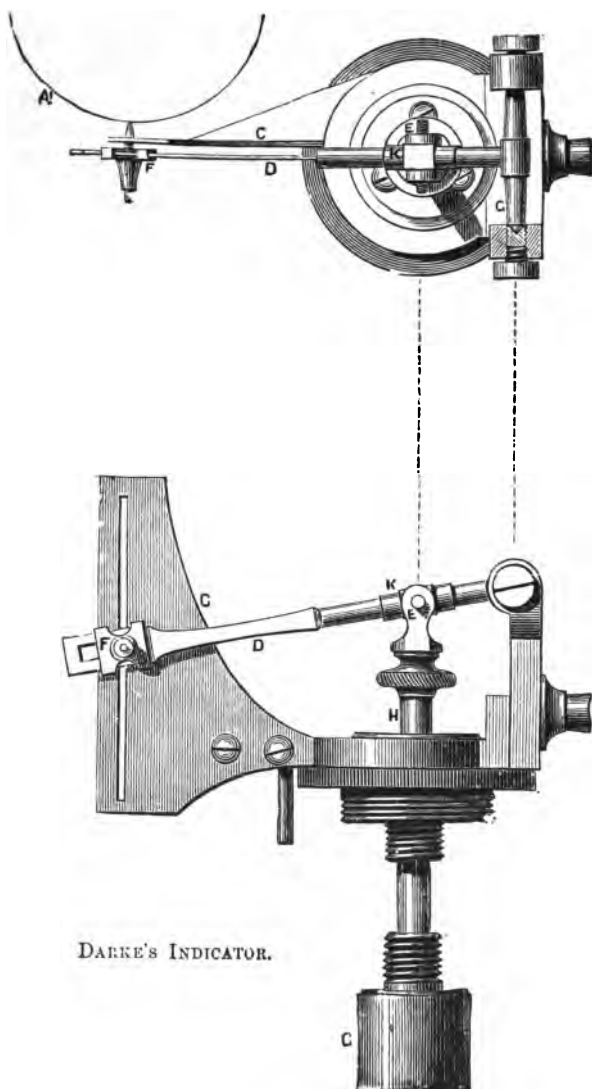


FIG. 14.

straight line parallel with the axis of the paper drum by being made to pass through a slot guide in a plate, C. The piston-rod head carries a jaw, E, and in this is a sleeve, K (Figs. 15, 16), which is an accurate but easy fit on the pencil arm, which is made round at this part. With the motion of the piston and



FIGS. 15 AND 16.

the jaw E, the sleeve K is caused to slide through a small range on the pencil arm as the latter describes an arc on its pivots. The end of the pencil arm, also, in a similar way slides in the piece F, which is compelled by the slot to keep the straight line of the slot. It may be here mentioned that a simple light pencil arm guided by a vertical slot was made by Mr. Richardson, of Messrs. Elliott's, in 1878. Our engraving shows Darke's indicator full size. The piston rod is hollow, see Fig. 17, the piston is made smaller and of shorter stroke than in the Richards, and there are differences in the springs and also in the construction of the jaw head, seen in Fig. 17, which we need not describe, as the forms adopted are so well shown in our engravings. In Figs. 13 and 14, Darke's detent gear, which we described in our last impression, is seen at M L, the pawl M being out of gear in Fig. 14; but a touch of the knob on the spring L will cause it to engage in the teeth at N (Fig. 14) when the drum is pulled round by the string. By this means the drum is stopped without disconnecting the string from the engine.

The piston spring is screwed to the screw on the piston G (Fig. 16), and to the screw projecting downwardly from the cover, as also seen in the same detached detail drawing A. As shown in our engravings, the drum is made to contain a roll of diagram paper, A', that part which is outside the drum being held in place by the clip springs O O. When a diagram is taken the paper containing it is torn off, the removal in this way bringing forward and placing paper ready for the next diagram. This arrangement is used in most of the Darke indicators, but engineers preferring to put on separate papers may have the drum fitted for this. This indicator is, as we previously remarked, made by Messrs. Elliott Brothers, Charing-cross, and the value of the improvements it contains

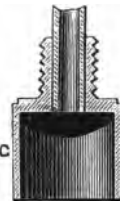
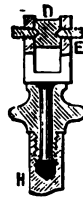
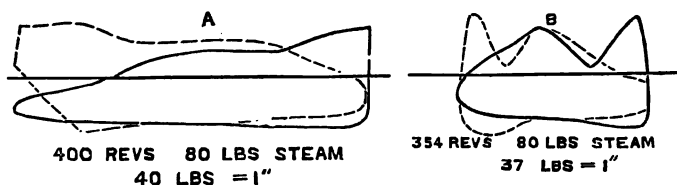


FIG. 17.

may be seen from the diagrams (Figs. 18 and 19). The diagram marked B is taken by a Richards indicator from the low-pressure cylinder of a torpedo boat engine, and that marked A from engines of similar construction working under similar conditions, but taken with a Darke's indicator. The torpedo boat engines were running at several hundred revolutions per minute. These diagrams show the very great and misleading effect obtained with the high speed engine as due to the momentum of the moving parts of a *Richards indicator*. The difference in the diagram is equally evident from slower running engines with quick cut off, as in the Corliss engine.

There is yet another indicator which, by reason of its good qualities, and especially of its steadiness in working at high



FIGS. 18 AND 19.

speeds, has claimed a good deal of attention amongst English mechanics during the past year or two, namely, the indicator made by the Crosby Steam Gauge and Valve Company, of Boston, U.S., a company whose London office is in Cross-street, Finsbury. This indicator we must illustrate and describe, as it affords a good example of that capacity for the development of an invention which seems to characterise American inventors of a large class, and it is noteworthy that the indicator which has held its own for so many years, viz., the Richards, was also an American development of the English invention.

The Crosby indicator is illustrated in perspective and in section by Figs. 20 and 21. Both these views show the lightness of the pencil motion, the general simplicity of the whole indicator, the simple arrangement of piston, piston-rod,

and spring. The spring differs in one feature from those used in any previous indicator, namely, that its lower end is not rigidly fixed by a screwed collar on the piston-rod, but the spring itself is a double twist of one piece of steel wire, to the middle of which is attached a small steel sphere, which sits in the hollow piston rod, as shown in Fig. 21. so that the spring is free to adjust itself, and to allow the piston and rod to move freely and without the slight friction which want of absolute centralism of the internal and external screws of the usual arrangement sometimes cause. The Crosby spring, seen at full size in Fig. 22, is screwed rigidly to the top cover, as in other indicators, so it needs to be as accurately made in this respect; but the freedom given at the piston end is undoubtedly of advantage, while the lightness of that end is of much greater importance, as this part receives the same motion as the piston.

Turning to Fig. 22, it will be observed that the head by which the spring is held to the cover A (Fig. 21) is large and heavy. Here, however, size is of great advantage, for it secures the advantages of a method of holding that end of the spring, which enables the manufacturers to adjust the scale of the spring with great facility. In the ordinary way it is customary to effect the final adjustment of the scale of springs, or, rather, to give them the necessary range with a given amount of pressure, by grinding the wire of the spring after it has been tempered. This, it need hardly be said, is not a simple operation, and to some extent affects the uniformity of the sectional area of the spring wire, and therefore detracts from the desired uniformity of flexure throughout the length of the spring. In the Crosby spring, however, the two ends of the wire are (the spring being like a double spring) screwed or threaded into the holes, drilled on the wings of the head, and all that need be done to adjust their scale is to screw them in or out of this head. During compression, spiral springs often compress round an axis which is not either a perfectly right line or a fixed one. The advantage of the freedom given by the ball joint connection of the piston and

spring is therefore of much importance, considering that it is the attention to these many small things, which makes up the superiority of the modern indicators. It will be seen from Fig. 21 that the joint part E of the piston rod screws into the upper part of the tube C, which, at its lower end, screws into the upper part of the boss of the piston H. The screw G in the

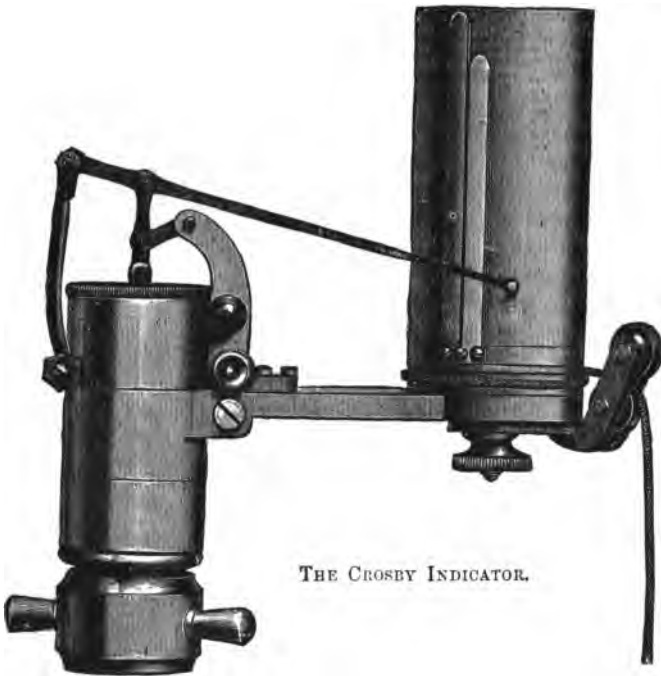


FIG. 20.

piston is to adjust the grip on the ball joint of the spring. At F round the piston are small chambers into which steam finds its way to cause the piston to maintain a central position in the cylinder.

Turning to the paper drum, seen in section in Fig. 21, it will be observed that round the central spindle is a spiral

spring. The upper end of this spring is fixed by the nut and collar at the top of the spindle, the lower end being attached to the bottom part of the drum. By loosening the upper nut and turning the collar either the one or the other way, and then setting it fast by the nut, greater or less tension may with facility be put upon the spring to suit the speed at which

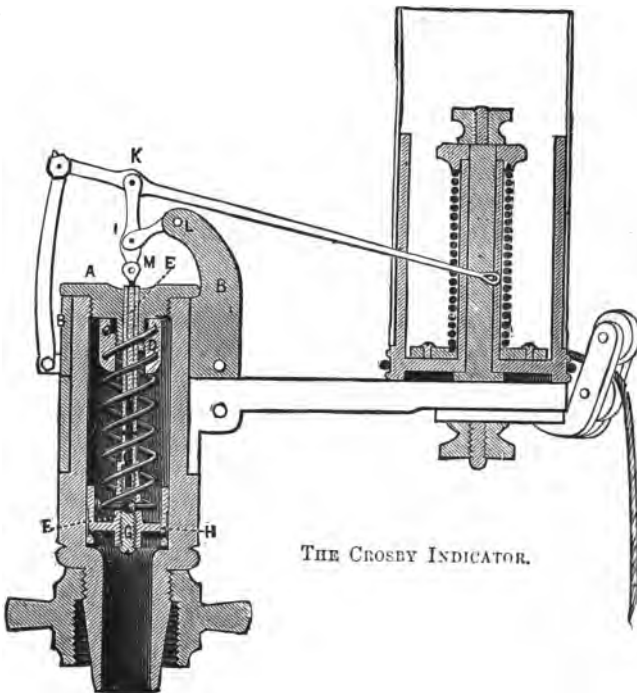


FIG. 21.

the engine to be indicated is running. This spiral spring does not give a uniform tension throughout the rotation of the drum. It happens, however, that the increasing resistance of the spring as the cord pulls it to the end of the stroke, is exactly what is required for high speed indicating. For instance, at the end of one and the commencement of another

stroke the indicator drum is at rest for an assignable length of time whatever the speed of the engine. From rest it is jerked into motion, in some cases of very high velocity. When this is the case the string has considerable work to do in overcoming the inertia of the drum. At this period the spring should offer only sufficient resistance to the movement of the drum to prevent the drum from making any back lash movement at the first jerk of the string. On the other hand, at the end of the out stroke of the string, the tension may and should be higher; the string would be able to stand the higher pull, as it and the drum will have been falling in speed as the steam engine is finishing its stroke, and the higher tension of the wound up spring will enable it to the better overcome the inertia and start the drum on its backward rotation.



FIG. 22.

A very interesting illustration of the reality of the varying stress upon an indicator string, and therefore of the necessity of an oppositely varying tension on the drum spring, is given by the Crosby Company, as obtained by means of an instrument, the invention of Mr. G. W. Brown. This instrument is illustrated by Fig. 23, and diagrams taken by its aid, showing the variation in the tension on the string, are given in Figs. 24 to 27. If a perfectly inextensible string could be obtained, the difference of tension would make no difference in the steam diagram, and even with the good string ordinarily used there is barely any appreciable effect on the diagram. The indications are, nevertheless, more likely to be accurate when the whole indicator works smoothly than when otherwise, and the results obtained by the instrument shown at Fig. 23 are in themselves interesting. The drum spring testing instrument consists of a wooden plate, A, to one end of which is fastened the brass frame BB, carrying the slide C with its cross head D. The head of the spring F is screwed to the cross head, while the other end is connected with the

bent lever G, carrying the pencil. The connecting rod E, which moves the slide C, receives its motion from a crank not shown in the illustration. The swinging leaf holds the paper on which the diagram is to be taken. The indicator to be tested is clamped to the plate as shown, and the drum cord connected with the free end of the spring. The crank is made to move at the speed at which it is desired to test the drum and spring. The paper is then pressed up to the pencil and the diagram taken. If the stress on the cord is constant, the lines which represent the forward and return strokes will be parallel to the motion of the slide, but, if the stress is not constant the pencil will rise and fall as the strain is greater or less. The line below the diagram is the line of no stress, and is drawn when the cord has been detached from the indicator. In the diagram, horizontal distance represents the position of the drum, and vertical distance represents the strain on the cord. The perfect diagram would be two lines near together and parallel to the line of no stress, and would represent a constant stress, and consequently a constant stretch of the cord, from which no error would result.

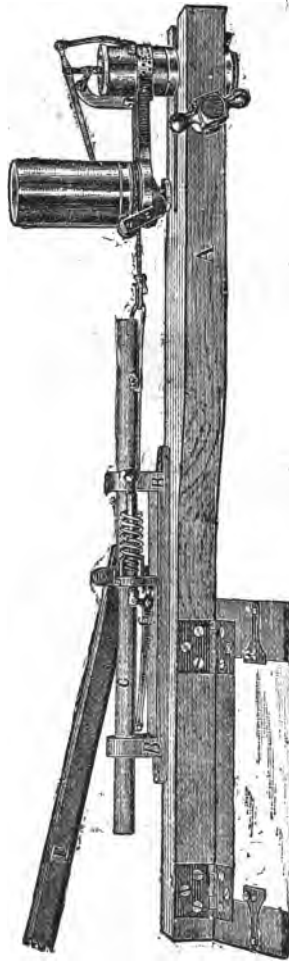


FIG. 23.

The diagrams, Figs. 24 to 27, are shown two-thirds their original size.

Fig. 24 shows how, with a Thompson indicator drum, the



FIG. 24.

stress on the string at 250 revolutions per minute fluctuated from but a slight one to a high one, falling again before the



FIG. 25.

end of the stroke, when it rose to the point from which it fell rose and fell to the starting point. Fig. 25 shows the stresses,



FIG. 26.

as obtained by Messrs. Crosby with their own drum spring; here the stresses become so near as to be almost a single



FIG. 27.

stress Fig. 26, also obtained by the makers themselves, is from a Tabor indicator drum at 400 revolutions per minute,

while Fig. 27 is from a Crosby drum at 450 revolutions. In both Figs. 24 and 26 the straight dotted line is merely to connect the two points at the ends of the stroke, to show the departure of the stresses from a uniform rise and fall with the stroke. These diagrams have an interest from a mechanical point of view apart from indicators themselves. They serve to show how varied may be the stresses where uniformity, or approximately uniform rise or fall, might be expected.

The guide pulleys seen in Figs. 20 and 21 are adjustable as to position by means of the nut below the drum, so that the string may be led in any desired direction.

The spring clips on the Crosby drum are, it will be seen, not of equal length. This is an apparently small invention, but it really is one which saves a great deal of trouble. It makes the application of the indicator paper very much easier and quicker than as ordinarily made.

The Thompson and the Tabor indicators referred to are two indicators which have been a good deal used in the United States, although they have been little known in England. One of them, the Thompson, is very much like the Kraft indicator; it has very light pencil gear, the pencil arm having a free end.

Numerous other forms of indicators have been made, and have been more or less common on the Continent, including the Rosenkranz, the Martin-Garnier, the Schaeffer and Budenberg, and the Deprez. In England the Hopkinson indicator was a good deal used; it had a very light pencil gear, but the range of movement was only that of the spring. The Casartelli indicator has also been a good deal used; its chief distinguishing feature is a light parallel motion for amplifying the range of movement of the spring or piston rod. The Kenyon indicator has no piston, the motion being imparted to the pencil by the flexure of a bent tube like that of a Bourdon pressure gauge. For a description of these and others reference may be made to the "*Guide pour l'Essai des Machines à Vapeur*," by M. J. Buchetti.

From the descriptions we have given of indicators it will have been gathered that the aim of inventors has been to

overcome the effects of inertia of those parts of the indicator which have to be put into rapid motion and to be brought quickly to rest. This has involved the use of very light parts for the pencil-carrying mechanism, and the problem of imparting sufficient strength to these parts has been well solved, particularly in the Crosby and Darke indicators. It has also become necessary in all cases to amplify the range of compression of the spring for the movement of the pencil, in order that the spring may be as short as possible. In the Crosby indicator this amplification is about six times. Thus, for 1 inch of movement of the indicator pencil only about one-sixth of the compression of the spring necessary in the Watt, the McNaught or the Hopkinson indicators, as referred to at p. 15, would be required. The spring is thus reduced in length, in weight, in inertia and in elastic vibration.



PART II.

ATTACHMENT OF THE INDICATOR AND APPARATUS FOR ACTUATING IT.

ASSUMING that we have sufficiently described and illustrated the modern steam engine indicator, and given a general idea of the nature and purpose of the indicator diagram, we may now proceed to consider the application and employment of the indicator and the use in particular of the diagrams obtained by it.

We have, then, first to attach the indicator to the steam engine cylinder. This must be done so that the supply of steam to the indicator is perfectly free—that is to say, so that the pressure in the cylinder is faithfully represented by that on the indicator piston. To effect this the passage from the cylinder must in no way be obstructed either by the use of connections with bore smaller than that of the indicator cock, and in all cases where a nipple, socket, or pipe is required they should be larger than this. No harm can be done by having such fittings or connections of from half-inch to three-quarter inch bore. When possible, the indicator cock should be screwed direct into the cylinder or the cylinder cover. All good engine cylinders

are provided with a boss at the cylinder ends, or a local thickening of the cylinder flange, as shown at B, Fig. 28, the flange at other parts being as seen at Fig. 29. In these sketches A is part of a horizontal engine cylinder, C the cover, P the steam port, E the slightly enlarged end of the cylinder by which clearance is given to the piston, the latter just passing the edge E, and thus the whole of that part of the cylinder

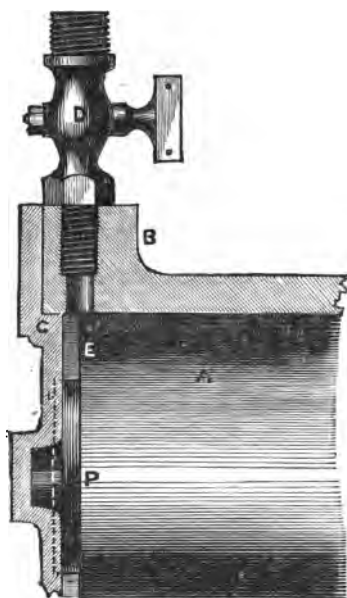


FIG. 28.

which is of the diameter of the piston is swept through, and wear does not leave a rim at the ends of the original diameter of the cylinder which may do harm, when, by fitting the connecting rod with new brasses, the piston is made to sweep over the rim left as the connecting rod is shortened up by wear of the old brasses. The boss B, then, is screwed with a three-quarter inch Whitworth thread, this being the size usually adopted for the indicator cock D, the upper part of which is

internally coned to fit the cone at the end of the indicator cylinder, as already shown at, for instance, Fig. 21. The lower part of the hole in the boss B should be half an inch in diameter, and should be as near the end of the cylinder as possible, so that it is not covered by the steam engine piston, but at the same time it should be clear of the cover C.

Indicator diagrams should always be taken from both ends of a cylinder, as, although it may be supposed that the slide valve is accurately set, this is not necessarily the case. Moreover, the valve itself may not be properly proportioned, and the cut off and lead may alter with wear of the excentric and straps and the rod joint. There may be obstructions in one

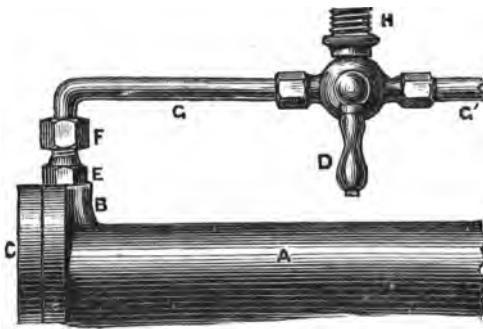


FIG. 29.

of the ports, the exhaust may not be so free from one end as the other, owing to misconstruction of the valve and the wear already mentioned. All these things would be found out by taking diagrams from both ends of the cylinder. For this purpose the boss (Figs. 28 and 29) should be made. When it is desired to take diagrams at frequent intervals the two ends of the cylinder may be connected with a pipe, G, as shown at Fig. 29, which is one of several forms of such connections. A short piece, E, in the place of the cock shown at Fig. 28, is screwed into the cylinder, the upper part of this being screwed to take the union nut F, which is fitted to the end of the pipe G; the bend in the pipe should not be of small

radius, although incorrectly shown with a short bend in the engraving, and should never be replaced by a right-angled elbow, as sudden bends have some effect on the flow of the steam, and may prevent the pressure in the indicator from instantaneously acquiring that which exists in the engine cylinder, and thus give a slanting instead of vertical line at the steam end of the diagram. At D is a three way cock, by means of which the indicator is supported at H, and the steam admitted from either the one or the other end of the cylinder by the pipe G or G¹. The plug of the cock D should be made larger than would be sufficient for ordinary work, so that the hole through it should not only be large enough to pass steam

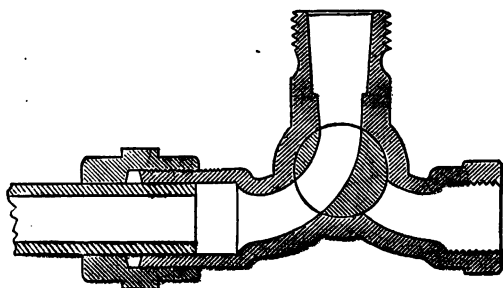
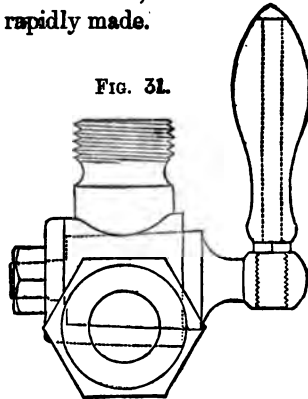


FIG. 33.

freely, but so that the hole may be curved as shown at Fig. 30, which is a section of the three-way cock (Fig. 31) as made by the Crosby Steam-Gauge Valve Company for use with the Crosby indicator. It will be observed that the steam passes round an easy curve in this cock whether the plug be turned towards one or the other way. At one end of this cock is shown an expansion joint (Fig. 30), which for the temporary application of an indicator to different engines gives facility in connecting at and allows for expansion of the fittings.

When this arrangement is adopted, the diagrams from both ends of the cylinder may be taken on the same cards as shown at Fig. 32, by simply turning the cock D, so that steam is alternately admitted from either end of the cylinder. This is

of some advantage when diagrams have frequently to be taken, as the two diagrams are correctly placed in a comparative position, and a picture of what is going on during the strokes in both directions is obtained, from which a judgment of the working can be rapidly made.



Where, however, only a few occasional diagrams have to be taken, and accurate measurements to be obtained from them, instead of simply obtaining a general diagnosis of the performance of the engine as far as the behaviour of the steam in the cylinder is concerned, the separate diagrams are to be preferred. When large engines have to be indicated it is pre-

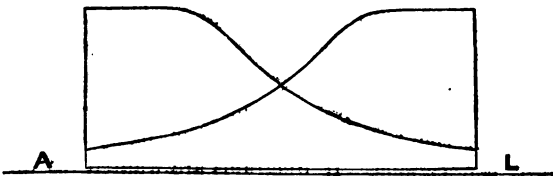


FIG. 32.

ferable that two indicators should be used rather than employing long pipe connections between the two ends of the cylinder. Steam in long pipes is likely to condense, and the presence of water in the pipes or in the indicator will destroy the truth of the diagram. The full pressure of the steam in the cylinder is not shown at the commencement of the stroke on the dia-

gram, and the steam line is made irregular by the action of the water in pipe connections. In order to take two diagrams at nearly the same time, it is therefore necessary to employ an indicator at each end of the cylinder. For the purposes, however, of a general indication of the working of an engine on continuous work, as in flour mills, and its power, one indicator moved from time to time may be sufficient.

When vertical engines have to be indicated, an arrangement similar to that shown at Fig. 29 may be employed. This would hold the indicator in a horizontal position, which is in some respects undesirable, and may be avoided by having a bend fitted to the three-way cock, with a socket into which the indicator cock may be screwed. When this arrangement is not used, the indicator may be screwed direct into the top cylinder cover for the upper end of the cylinder, and for the lower end a pipe with bends must be used, so as to bring the indicator clear of the cylinder, and in a vertical position. The position for fixing must always be chosen with reference to the part of the engine from which motion is obtained for the indicator drum. When the pipe connections are not so attached that water may easily drain away from them, they should be provided with a small cock by which it may be allowed to escape. Provision for this should always be made. In all cases care should be taken that these connections are properly and firmly made and steam tight, and they should be strong, so that the indicator may be firmly held. It costs little more to do the work well, and diagrams upon which reliance may be placed cannot be obtained with the indicator hastily and badly applied with makeshift connections. If diagrams are wanted at all they should be accurate, and if it is worth taking them, it is worth taking them well.

The mode of working the indicator has now to be considered—that is to say, the means by which a movement of the indicator barrel exactly synchronous with that of the steam engine piston is obtained. This is not so simple a matter where exactness is required as is usually supposed, and although it

is not desirable to insist upon mathematical precision and perfection of fitting of indicator gear, and on the necessity for perfect inextensibility in the communicating cord, it is essential that some thought and care should be bestowed on these matters. It is especially necessary that the indicator card should move in as perfect unison with the piston as though it were attached to it. To obtain this, the gear by which the stroke of the piston is converted into the shorter one of the indicator card should be rigid but light in construction, and the best cord that is made should be used to connect the gear with the indicator barrel. Some engineers prefer to use fine hard brass or steel wire for this purpose. In order to make clear the inaccuracies which accrue in the indicator diagram, through imperfection of the reducing and communicating gear and cord, it is necessary in the first place to describe the arrangements of gear more commonly employed. We will assume that we are dealing with a horizontal engine, although the gear we shall describe will, as shown, or with very slight addition or modification, be equally applicable to vertical engines. With few exceptions the horizontal and the vertical include all types, though the multiplication of levers and rods in marine engines sometimes makes it difficult in the usually confined space of an engine room to fit up satisfactory indicator gear. The gear necessary for working an indicator on an oscillating cylinder engine is of a different character, and these diagrams are more difficult to take, in consequence chiefly of the movement of the cylinder.

In Fig. 33 we have a diagram illustrative of a very usual form of reducing gear. Attached to the bolts that hold the guide bars in place are the legs *D* of a stiff standard for holding the spindle or the pivot *p* upon which the swinging arm *A* is pivoted. *C* is the engine crosshead, and in this a pin is fixed for communicating the movement of the piston to the arm *A*, the difference between the circular path *xx* of a point in *A*, and the straight path of the crosshead *C*, being accommodated by the slot in the end of *A*. In some cases the lever *A* is made of wood, but it can be much more satisfactorily made

in metal. The lower end of the arm A moves through a distance equal to the stroke of the piston, and a synchronous movement of shorter range, it is obvious, may be imparted to anything, say an indicator, connected by cord to the arm A at any point, y (not shown), intermediate between C and p . The point y may, for the sake of fixing ideas, be supposed to be at the lower of the two screws by which the wood segment B is attached to the lever A. The relation between the ranges of motion of indicator card

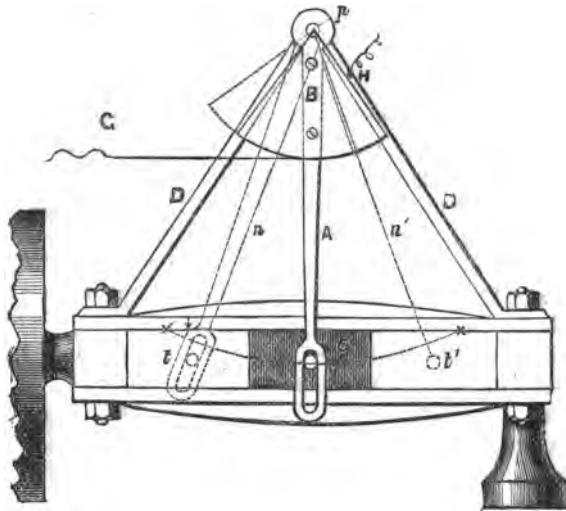


FIG. 33.

and of the piston will of course be that of the length pC to py . Ordinarily, the cord or wire G by which A communicates movement to the indicator barrel is attached to a pin screwed into A or into a piece of wood fastened thereto for the purpose. But the practice is becoming more general to attach a piece of wood of the segmental form shown at B, the radius of which is such as to give the desired length of stroke to the indicator card. Thus, suppose the stroke of the engine to be indicated be 30in., the length pC , say 36in., and it is

desired to take diagrams which shall be 4in. in length, the radius of the segment B, or the distance from *p* of a pin for the cord must be $r = \frac{\text{length of arm A}}{\text{stroke of engine} \div \text{stroke of card}}$, or we

have $r = \frac{\text{length of arm A} \times \text{stroke of card}}{\text{stroke of engine piston}}$, giving, in the case

assumed, a radius of $\frac{36 \times 4}{30} = 4.8\text{in.}$

The advantage which attends the use of the segment is that as the cord is always at the same vertical distance from *p* there is not that loss of horizontal movement in the cord due to the difference between the lengths of the chord and of the arc described, which would result from its following the angular movement of a pin at distance from *p* = the assumed radius *r*. The difference between the chord and the arc of the path of the pin is not great, but whether the indicator be near or far from the reducing gear it is sufficient to make an appreciable and sometimes important difference in the diagram. When the gear is very close to the indicator barrel the difference may be magnified. It may, however, be noted that as the pin in the crosshead C moves in its straight path, and the arm A moves in an angular path to *n*, the pin in the crosshead takes a new place in the slot; it is at a distance from *p*, which is greater than when the arm A is in midstroke, as shown, by the distance between the pin *t* and the arc *x x*. The length of the arm A is thus virtually increased, the relation between *p C* and *p r* no longer remains the same, and the result is that the piston of the engine and the crosshead C move through a horizontal distance which is greater than that communicated to that part of A where the pin stood at midstroke, by half the difference between the length of the chord of the arc described by it and of the stroke *tt'*. The reality of this movement of the piston without proper corresponding movement of the arm may be easily seen by imagining the arm A pivoted at, say, one-fourth the distance above C, and the slot made of the extra necessary length. This difference it will be seen tends to aggravate the errors

produced at either end of the diagram. The error is lessened in very rapid ratio with increase in length in the arm A. As far as the error is due to string attachment to a pin in A at the necessary distance from p , this is overcome by the use of the segment, which must be specially made for each case, as shown; and to avoid the error as caused by the use of the slotted lever, shown at Fig. 33, the arrangement shown at Fig. 34 is generally preferred. Here the loss of movement is only that due to the angular movement of the connecting rod

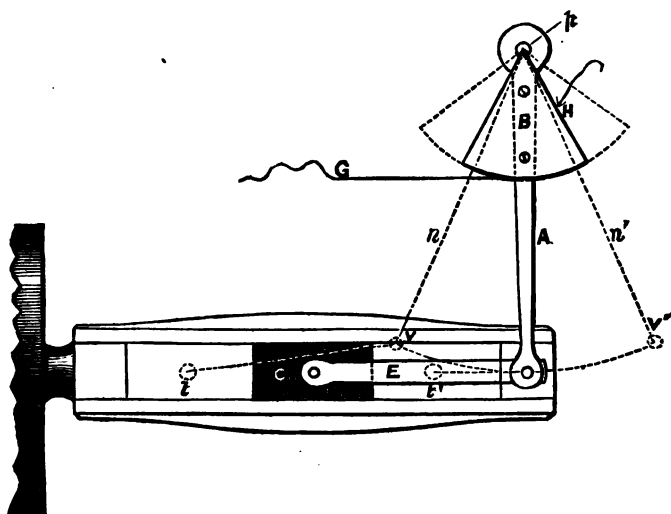


FIG. 34.

E, and is equal to the versed sine of the arc described by it. This, it will be seen may be made inappreciable by giving the rods E and A considerable length. It will be observed that this loss of movement or inaccuracy at the two ends of the cord happens when the engine piston is moving most slowly, and it becomes an important matter when, as in the case more especially of gas engines, it is desired to find the time occupied in acquiring the maximum pressure. In Fig. 34 the mode of supporting the pivot of the arm A is not shown; but this,

as in the case of Fig. 33, may be of any form most easily made, provided it be rigid and the bearings and pins properly fitted. The cord or string G is attached to a pin in the side of the segment at H. By using the segment B the string G may be led to the indicator in whatever position in a vertical plane it may happen to be without affecting the proper movement of the card, the segment having the necessary length of arc, and being attached to A in a suitable position.

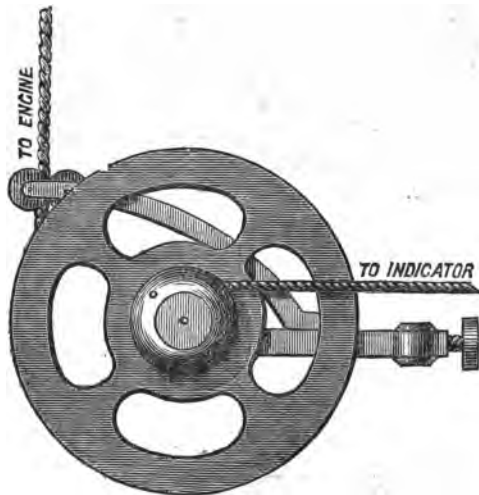


FIG. 35.

The segment is shown with its centre line coincident with that of A, but it need not, of course, have this position. Instead of a segment a circular disc may be used. In all cases care must be taken that the string G is of the proper length, so that the indicator barrel is not pulled to or let back to either of the stops.

When a segment or a disc is not used, but only a pin, as above mentioned, to which to attach the string G, care must be taken that a line connecting the pivot *p* and the pin be

normal to the indicator string when the piston is at half stroke, as it is in the diagrams (Figs. 33 and 34).

A form of reducing gear made by Messrs. Elliott Brothers is illustrated by Fig. 35. By means of this there are none of the inaccuracies due to what we may term angular causes already explained, the reduction being effected by the use of rollers or pulleys, the radii of which bear the same relation to each other which must be borne by the arm A and distance r already given. The apparatus necessitates different wheels for different strokes and lengths of diagram. It may be attached direct to the indicator, or elsewhere if more convenient, and is a handy form for oscillating cylinder engines. It necessitates the use of two strings or wires instead of one, and thus, although it avoids all the levers and supports we have described, it incurs whatever objections may be made to strings or wire.



PART III.

ENGINE DIAGRAMS, NON-COMPOUND.

WE may now assume that we are prepared to take an indicator diagram, and to gather from it the indicated power of the engine from which it is taken. Although this is but a small part of the purpose and value of an indicator diagram, we will deal with it first. Later on we shall consider how the information obtained from the indicator can be made available for understanding the behaviour of steam in the cylinder, and for interpreting the peculiarities which determine the efficiency of the steam engine, and the proper design and adjustment of its parts.

We will take as a first example a diagram (Fig. 36) made five years since from one of the last, if not the last, remaining atmospheric engine, and one which, therefore, is of some historic interest. The engine was apparently a Smeaton atmospheric engine, although it had subsequently been improved in some details. It was at work at the Handley Wood coal pit of the Staveley Company until November, 1879, and the date on its cylinder was 1776. It had thus been in existence and at work for the greater part of a century. At the time of its removal an engraving with sections and description of this heavy but hale old relic was published in *The Engineer*, from which this typical diagram of an engine of the low pressure type—a type that did good work—has been copied, and provides us with a fit primary example. The engine had a cylinder 48in.

diameter, and therefore of 1809.56 square inches area. The stroke was 5.4ft., and during the time the diagram (Fig. 36) was taken the engine made 19 strokes in three minutes, or 6.33 per minute. The steam was supplied by two egg-ended boilers, in which the pressure was about 14lb. per square inch only a few feet from the engine, but, as will be seen, the maximum pressure above the atmosphere was about 2.30lbs. per square inch. The necessary pressure was not great, as all that was done in the steam stroke was to get the piston to the top of the cylinder, and the pumps it worked, in position for the down or working stroke, effected by atmospheric pressure. The diagram has been divided into ten equal parts, and the mean pressure in each part marked therein. From the diagram it will be seen

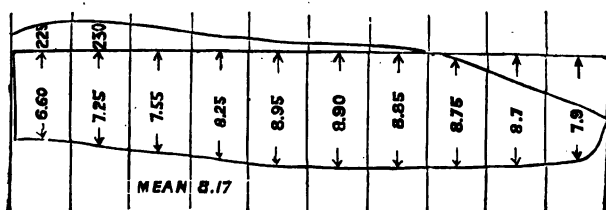


FIG. 36.

that the steam never rose in the up stroke of the piston above 2.3lbs., cylinder condensation being enormous, while in the first part of this stroke it did not rise even to this. As the piston rose the pressure fell, until, when the piston was at the top of its stroke, exhaust having taken place, the formation of a vacuum had commenced, and in the first tenth of the down stroke had reached 7.9lbs. below the atmosphere, so that the atmospheric pressure was made use of to that extent. This pressure grew, or in other words the vacuum improved, until about mid-stroke it reached 8.95lbs., from which point it fell to 6.6lbs. at the end of the stroke. In this case it is only the vacuum or atmospheric pressure which is credited with doing the work, as the down stroke of the piston is the stroke which raises the water or in which useful work is done, the

engine and pump being so nearly balanced that only a very slight pressure should be necessary to raise the piston. The pressure shown above the atmospheric line A L is therefore the friction part of the diagram, and shows the amount of power necessary simply to put the engine and pumps into motion. The ten pressures below atmospheric pressure shown by this diagram, which was taken by a 10lb. spring, and thus showed 1lb. pressure for each 0·10in., give a mean of 8·17lb. per square inch, so that the work done upon the piston by the atmosphere in the down stroke, that is in the effective working stroke, was

$$\frac{5\cdot4 \times 6\cdot33 \times 48^2 \times 0\cdot7854 \times 8\cdot17}{33000} = 15\cdot83,$$

indicated horse-power. The network represented by the water raised by the pumps was 10·05 horse-power, so that the effective work, was of the indicated work, as much as 63·6 per cent. Speaking of this result and of the engine, *The Engineer* remarked :—"Considering the condition of the engine, its disrepair (permitted because its days were numbered), the fact that a lin. water pipe had to be kept running, nearly full bore, into the cylinder, and about 1ft. of water maintained on the top of the piston, to keep it air-tight. . . . it may be considered that the old engine did not do so badly." Here it must be remarked that the up-stroke of the piston is what is called in pumping engine house language the outdoor stroke, the down stroke bringing the pump bucket up or towards "indoors." This will make what follows clear. After the above diagram was taken "the piston was repacked with a view to lessening the condensation of steam during the outdoor stroke. The hemp was taken out, and some packing with india rubber core put in. With this packing the condensation was less and the vacuum slightly improved, but the tightness of the piston in the cylinder so increased the friction that a much higher steam pressure was necessary to get it out of doors. In a few hours, however, it wore easier, and diagrams, of which the annexed Fig. 37 is an example, were taken." This

diagram we give as affording an excellent example of what may result simply from extra friction caused by tight packing, and of the way in which an increased indicated horse-power may be accompanied by no increase of actual power or of work done. From Fig. 37 "it is seen that with only the same boiler pressure, the pressure in the cylinder is nearly double what it was when so much water passed by the piston during its up stroke. The vacuum is improved, so that the mean effective atmospheric pressure, shown by the indicator diagram, was 9.15lb., the indication of the vacuum gauge reaching 19.5in. Even under these conditions, however, only 17.5 strokes were made per three minutes, or 5.83 per minute.

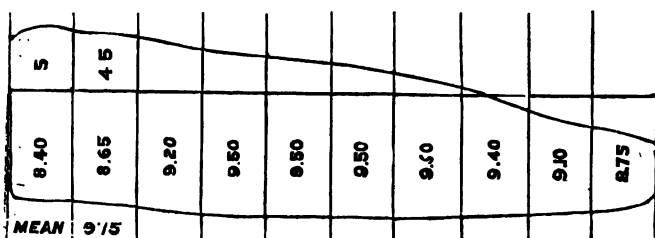


FIG. 37.

The work done on the piston was in this case, then,

$$\frac{5.41 \times 5.83 \times 1809.5 \times 9.15}{33000} = 15.80 \text{ horse-power, or practically}$$

the same as before, a result which might have been expected from the recorded experience of the use of atmospheric engines in days gone by, when it was often noticed that what might be called a considerable leakage past the piston was rather beneficial than otherwise." This, it must be observed, was not because the admission of water into the cylinder was itself advantageous, for, on the contrary, it was very wasteful, but it did not cause so much loss as a tight packing in a cylinder that was perhaps an inch out of truth in a length of three or four feet, and very rough also.

These diagrams will afford some idea of the way in which

the indicator served to throw light on the work done by atmospheric engines, for, with only about 15 indicated horse-power from the old engine referred to, in its dilapidated state, about 800lbs. of coal were burned per hour, or about 50lbs. per indicated horse-power, representing, with the boilers in use, about 300lbs. of steam per indicated horse-power, or more than twelve times as much as an ordinarily good high-pressure engine of modern make would require. Thus the indicator diagram without other information would show very little. These diagrams might be supposed to show very good performances, and without any knowledge of their history would be considered fairly good, although it is clear from them that from whatever engine, the vacuum is not as good as it should be and that towards the end of the stroke the vacuum decreases instead of being maintained at its maximum. In Fig. 36 it falls from 8·95 to 6·60, indicating a heavy leakage past piston or valves or in the condenser somewhere.

In other words, the diagrams, without full information concerning the engine from which they are taken, show very little more than pressure and its variation, and the difference between Figs. 36 and 37 would indicate difference in external load upon the engine, which is not true, as the difference is only that due to internal friction. With the knowledge of the engine and its condition of working, we see that steam pressure, which should not, and would not if the engine were in first-rate order, exceed a few ounces in the up stroke, reaches several pounds; we know that within a fraction all the work done, as indicated by that part of the diagram which is above the atmospheric line, is lost work, inasmuch as the pressure on the up stroke of the piston is not conveyed to or used in working the pump, the connection between engine beam and pump being flexible and incapable of transmitting downward pressure to the pump rods. If the connection between engine beam and pump rods had been rigid, so that the pump rods could have been lightened and required forcing down, and required, therefore, so much less to lift them; or if the beam had been loaded with a

weight which in the indoor stroke would have acted in concert with the atmospheric pressure, then steam pressure in the top of the diagram might have done useful work ; but this was not the case. The top of the diagram shows nothing but engine friction or lost work. If this top part of the diagram, or rather its amount, be deducted from the lower part of the diagram, the amount of external work done by the engine should be the result. A glance at the diagrams will show roughly that this is the case. The net work represented by the weight of water raised is very nearly two-thirds of the indicated work. If to this net work the friction of the water through the pipes and pump valves be taken into account, the diagrams would agree with the actual work. Or, on the other hand, if we reduce the lower part of the diagram to that which would represent the 10·05 horse-power above mentioned, and then deduct the amount of the top part of the diagram, we should have remaining a small balance of indicated power which would be the equivalent of the friction external to the engine and its connections, or the friction of the water in the pump pipes and valves.

It will be seen that in these diagrams the pressure in the up stroke falls below the atmospheric line, the inertia of motion of the engine and its connections being sufficient to perform the upper quarter of the stroke.

Our first actual diagrams have thus been from the earliest type of engine, engines nominally working with atmospheric pressure only. We shall next deal with diagrams from engines which do not make any use of the atmospheric pressure.

We may now take, as a simple example of a diagram from a high-pressure non-condensing engine, the one given, Fig. 38, which is from a portable engine by a good maker, but which had been at work several years, without any repairs or adjustment, when the diagram was taken. A thrashing machine was being driven, and the diagram was obtained when the machine was running empty, so that it indicates the power

employed simply to work the engine and machine without doing any thrashing.

The engine had a cylinder 8·5 in diameter and a stroke of 1ft. When the diagram was taken it was running at 115 revolutions per minute, and the boiler pressure was about 47lb. per square inch. The engine was fitted with a common slide valve, with lap sufficient to cut off the steam at about one-third of the stroke, and with a governor and common throttle valve which reduced the initial pressure in the cylinder to that shown on the diagram.

The diameter of the cylinder being 8·5in., the area of the piston was 56·74 square inches. The diagram (Fig. 38) is one of a large number taken during an experimental test of the machine; and it being necessary to take out the indicated

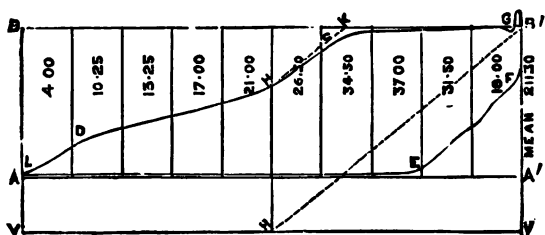


FIG. 38.

power as shown with the machine working under different conditions, the practice of using a constant to simplify the work of calculation, was adopted, and of this we may take an example. The constant for an engine is simply the number which represents the power of the engine per lb. pressure on the piston. In our case, then, it will be taking the same notation as in page 13 and C as representing engine constants,

$$C = \frac{A (S \times 2) R}{33,000} = \frac{56.74 \times 115 \times 2}{33,000} = 0.3954.$$

Our diagram gives us a mean pressure of 21·3lbs. pressure throughout the whole stroke. The indicated horse-power of the engine is then $0.3954 \times 21.3 = 8.42$. This, however, is only

as indicated by one end of the cylinder. The diagram from the other end showed how necessary it is to take diagrams from both ends of a cylinder and never to rely upon that from one end only. The adjustment of the slide valve being inaccurate the mean pressure was reduced to 18.3lbs., giving only 7.23 horse-power. The mean indicated horse-power was thus not 8.42 but $8.42 + 7.23 \div 2 = 7.82$.

In small engines the reduction in the effective area of the piston which is made by the piston rod is often—we might say generally—neglected. In larger engines it must be taken into consideration, and we may take this opportunity of an example of its effect. It will of course be very small in so small an engine, but not too small to give some idea of the necessity for making the reduction when the piston rods are large.

In our case the piston rod was 1.5in. diameter; this represents an area of 1.76 square inch, and by this area that of the piston will be reduced at one side, so that the effective area will be reduced by $1.76 \div 2 = 0.88$ in., and will equal $56.74 - 0.88$ in. = 55.86. Our engine constant will then be $C = \frac{55.86 \times 115 \times 2}{33,000} = 0.389$. Taking our pressure as the mean

of the pressures from the two ends of the cylinder, or 19.8lbs., we have the mean indicated horse-power of the engine $= 0.389 \times 198 = 7.70$, so that the piston rod occupies an area which reduces the power in this small engine by an eighth of a horse-power.

The diagram we have here used is not put forward as an example either of good diagrams or good steam performance, but is taken as illustrating what would be obtained from a very common type of engine, and it shows a good many points to which we must refer hereafter. In our engraving we have marked the atmospheric line A A', the line of maximum pressure at B' has been produced to B, so as to form with the vertical lines a parallelogram enclosing the diagram. In tracing the indicator pencil as it produced this diagram, we will commence at L—namely, at the beginning of the return

or exhaust-stroke. The diagram is here very near the atmospheric line, the pressure in the cylinder during the exhaust-stroke, or back pressure, being not more than 1lb. per square inch until it nears the last fourth of the stroke and the exhaust port is nearly closed. At E the exhaust port closes, and the vapour left in the cylinder is compressed by the piston as it completes its stroke. The pressure of this compressed exhaust vapour rises to F, or to about 26lb. per square inch. At F the steam port opens and steam is admitted while the piston has yet about one-sixteenth of an inch of its stroke to complete—that is to say, the valve is set with a sixteenth of an inch lead. The sudden admission of the steam drives the indicator pencil up to G above the normal maximum, and the spring in its recoil brings the pencil somewhat below this line. After this a clean line is drawn, the pencil falling gradually as the valve slowly cuts off the steam, the fall in pressure towards C being the result of the wire drawing, as it is termed, which is caused by this slow closing of the port. When a steam port is closed slowly, as it is by an ordinary slide valve without much lap, the pressure in the cylinder falls in consequence of the increase in the velocity of the piston which takes place as the port opening is being reduced to nothing, so that the passage gradually becomes too small to permit the steam to enter the cylinder at a rate sufficient to maintain the pressure behind the piston. A little before the pencil arrives at C steam is cut off and expansion begins, the fall in pressure being rapid at first between C and H, and from this point it is gradual, the slowness of the fall indicating re-evaporation of steam condensed during the previous part of the stroke. At O the exhaust port opens and the pressure falls to near that of the atmosphere at L when the stroke is completed.

Viewed with reference to economy this diagram shows too much compression for the speed of the piston, which is only 230ft. per minute; throttling, slow-cut off and wire-drawing, and unnecessarily high pressure at exhaust. Other points to be dealt with by reference to this diagram will be considered

hereafter, but before dealing with these and with diagrams from compound engines we will take a typical condensing engine diagram and a theoretical diagram into consideration.

Before proceeding to the condensing engine diagram, we will here give one from a higher type of high pressure non-condensing and, for the stroke, high speed engine, namely, an Allen engine. It is given in this place merely to show the form of diagram which results from long range expansion, or, in other words, an early cut off. Fig. 39 is from an engine with an 8in. cylinder, area = 50.26, stroke 2ft., revolutions 150 per minute, and steam cut off at one-tenth of the stroke, giving a ratio of expansion of ten. The engine

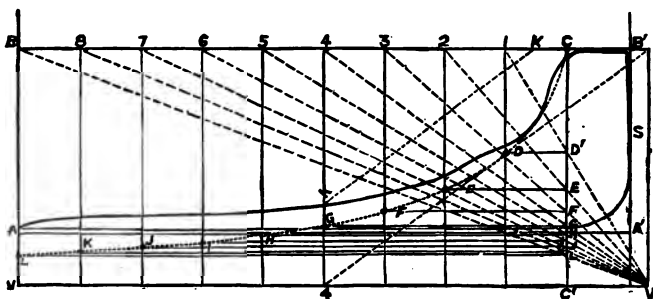


FIG. 39.

was running light at the time the diagram was taken with a Richards indicator, and the back pressure, which is shown by the space between the lower line of the diagram and the atmospheric line A A', was caused by the long length of exhaust pipe which was attached to the engine, aggravated by several bends. We have drawn an enclosing parallelogram about the diagram for the purpose of explanations, to which we shall recur later on. It will be seen that the steam admission line S departs from the vertical above S, showing that the full pressure was not realised until the piston had moved about a quarter of an inch, the admission being somewhat restricted, there being practically no lead. At C the cut off took place, that

is at the end of the first tenth of the stroke, and the pressure fell to one-half the maximum when the next tenth of the stroke was completed. From thence it falls gradually, until at the end of the stroke the pressure was 4.25lb. Exhaust then took place, but the pressure did not drop below 2.25lb., until the piston had made a considerable part of its return stroke, and never much below 2lb. At G the exhaust port is closed, and compression commenced. The valve gear of this engine is of a very quick acting kind, and cut off takes place without wire drawing. It will be remarked that the curve falls just above the letter D, forming a hollow curve. Between the division lines 1 and 2 the curve becomes convex, and afterwards again assumes the proper hollow expansion curve. The wavy line between or within the second and third divisions is caused by the oscillation of the indicator spring, and the inertia of the attached parts, both being brought into play by the sudden fall of pressure and correspondingly sudden movement of the indicator piston.

There are other causes assumed to act to bring about this wave line, which is often much more pronounced than is here shown from an engine running at 150 revolutions per minute. Water in the cylinder, the indicator, or connections, will cause this wavy line. We have not written upon this diagram (Fig. 39) the various pressures, as these figures would have confused an already filled diagram, but a measurement gives a mean pressure of 14.2lb. per square inch, the maximum pressure being 50lb. The piston rod being 1.75in. diameter gives an area of 2.4 square inches. As the rod only passed through one cover, we have to deduct only half this area from that of the piston, bringing it down from 50.26 to 49.06, or, say, 49 square inches. The indicated power of this engine was thus
$$\frac{49 \times 2 \times 150 \times 2 \times 14.2}{33,000} = 12.78 \text{ horse-}$$
power.

Or we may simplify the figures by taking the piston speed in feet per minute and obtain the constant C (page 61), or

the power per lb. pressure. The piston speed being 600, we have $C = \frac{600 \times 49}{33,000} = 0.899$, or, say, 0.9. The indicated power at the above pressure is then $14.2 \times .9 = 12.78$.

We have drawn upon this diagram the theoretical curve of expansion according to Boyle's or Marriotte's law. It should, perhaps, be always called Boyle's law, as by him it was discovered in 1662, and afterwards verified by Marriotte, and is to the effect that the pressure of a gas at constant temperature varies inversely, as the volume of the space it occupies—that is to say that if p and v represent pressure and volume, then $p v = a$ constant. This is true for gases such as air, but it is not true for steam; but upon this assumption, this so-called theoretic curve is drawn, and is very valuable for purpose of comparison. The curve is hyperbolic, and in practice is very nearly approached in consequence of conditions which come into play when steam is used to perform work expansively in a cylinder, as will be explained. For the present we may notice that our diagram (Fig. 39) shows pressures which from the middle of the second tenth part of the stroke are greater than the hyperbolic curve demands when the cut off is at C. This is in consequence of the re-evaporation of a very large quantity of steam, which is condensed in the first part of the stroke on its entrance into a cylinder cooled down to the temperature of steam at the end of the previous stroke, at about 4lb. The absolute pressure, as shown by the above diagram, at the end of the stroke, is 19lb., corresponding to a temperature of 225deg.; the temperature at the commencement of the stroke is that corresponding to a pressure of 76lb. absolute, or 308deg., the range of temperature being 83 degs.; but if the heat of the steam condensed in the commencement of the stroke did not provide the means of re-evaporation of the water in the cylinder, and of maintaining the temperature throughout the stroke, the pressure would fall at least as low as shown by the hyperbolic or isothermal curve in diagram Fig. 39, or to as low as 7lb.—really it would fall much lower. The temperature due to 7lb. pressure is

177 deg., or a range of 131 degs. The lowering of temperature of the cylinder consequent upon this range of temperature calls for the condensation of a large quantity of steam during the commencement of every stroke, and this being re-evaporated during the later part of the stroke under the falling pressure and temperature, gives the steam pressure which keeps the indicator diagram high above the hyperbolic curve. Before leaving this diagram it may be pointed out that during a large part of the stroke the pressure is not sufficient to overcome engine friction, so that for this reason alone, apart from others of equal importance, it is clear that a range of expansion of ten, when performed in a single cylinder, is not an economical one.

Leaving this high-pressure non-condensing engine diagram for the present, we must turn to typical diagrams from high-pressure condensing engines as distinguished from the low-pressure condensing engine diagrams (Figs. 36 and 37) taken from the atmospheric engine referred to at page 55.

First, we must deal with the diagrams from a Cornish engine. Not because these are often met with now, and at any time only as pumping engines, but because the Cornish engine diagram is peculiar; and these pages would be incomplete without a brief reference to this, the most important, form of single acting condensing engine. It must be premised that the steam only does direct work in the down stroke of these engines or on the top of the piston; but indirectly it does work by condensation on the lower side of the piston. The weight of the pump rods and buckets is sufficient to pull the engine beam down at the pump end, and, of course, up at the engine piston end. No steam is thus directly required to raise the piston. The working of the engine is as follows:—Steam is admitted to the top of the piston, and is cut off earlier or later according to the load; when the piston reaches the bottom of the stroke a valve affording communication between the two ends of the cylinder, and called the equilibrium valve, is opened, and the steam from the top of the cylinder is thus free to pass to the bottom. Equilibrium is thus established between

the top and the bottom of the piston, and the piston is lifted by the beam, all the steam passing to the under side of the piston as it leaves the upper side, and the pressure falling somewhat. When the piston is at the top of the stroke the equilibrium valve is closed, and a valve communicating between the lower side of the piston and the condenser is opened, the steam is condensed, and a vacuum formed. At the same time the steam valve is opened to the top of the piston, and the down stroke is performed by steam pressure at the top of the piston and by vacuum beneath it, or by atmospheric pressure added to the steam pressure acting on the top of the piston. The dia-

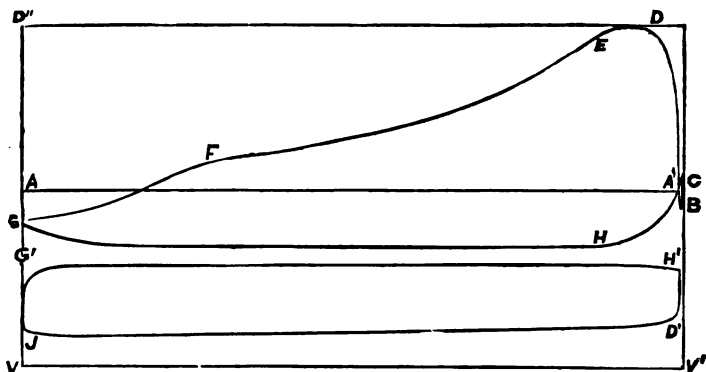


FIG. 40.

gram obtained from such an engine is given in Fig. 40. In this diagram, which is given full size as taken, the scale being one-sixteenth inch to the inch, A A' is the atmospheric pressure line, V V' the line of absolute vacuum. Tracing the pencil we may follow the piston in its downward stroke. Steam is admitted and the pencil rises from A A' to D, one-twentieth part of the stroke being performed before the maximum pressure is reached, owing to a somewhat restricted steam passage or slow valve. The piston descends, steam is cut off at somewhere near E, and expands down to F, when the equilibrium valve begins to open, and the pressure falls to G. At

this point the equilibrium valve is full open and the piston begins to rise, the steam passing from above it to below it as it does so, and the line $G H$ is traced from the top of the cylinder, while the line $G' H'$ is traced from the lower end, the gap between the two lines representing loss of pressure in the communications between the two ends of the cylinder. When the piston reaches the top of the stroke, not only is steam admitted, as before, to the top of the piston, but the exhaust valve opens

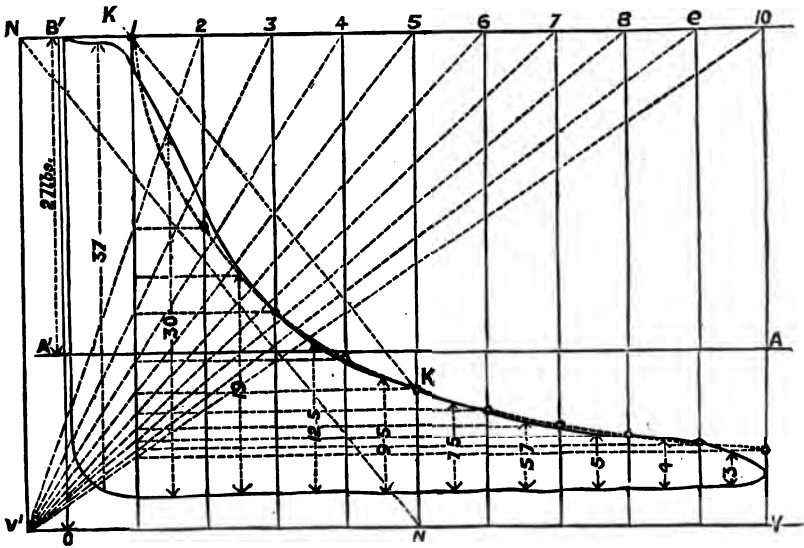


FIG. 41.

and the steam from below rushes into the condenser, and the pencil falls from H' to D' . The down stroke is now made under the pressure $D H + H' D'$; and (with an indicator at either end of the cylinder) the lines $A' D E F G$ and $D' J$ are simultaneously drawn. It will be noticed that in the up-stroke the equilibrium valve closes at H before the stroke is completed, and the steam imprisoned on the top of the cylinder is compressed, as shown by the curve $H C$. The compression

increases the pressure, and carries the pencil to C, but as soon as the piston stops, loss of heat by conduction, radiation, and leakage causes the fall of pressure again to B before the new steam enters, and hence the loop C B. When pressure has fallen to B, new steam enters, and the pencil rises as before to D, and the operations recommence.

We may now turn our attention to diagrams from high-pressure condensing engines working with moderate and high ratios of expansion. For this purpose we shall deal first with an engine working with a boiler pressure of 30lb., and a maximum cylinder pressure of 27lb., and with steam cut off at about one-tenth of the stroke, as shown by diagram (Fig. 41). The engine made 58 revolutions per minute, and with a mean pressure of 13·8lb. obtained from the diagram shown, and its fellow from the other end of the cylinder, indicated 463 horse-power. Fig. 41 gives a mean pressure, as will be seen from the figures upon it, of 13·27lb. By following this diagram it will be seen that the steam was admitted in such a way as to give full pressure in the cylinder instantaneously. It fell a little during the short period of admission, and at about one-tenth the stroke steam was fully cut off, and pressure fell to atmospheric pressure before four-tenths of the stroke had been completed, and continued to fall until a little below 3lb. per square inch. This would not usually be considered quite a sufficient terminal pressure, as the back pressure or imperfect vacuum represents rather more than 2lb. Deducting this back pressure from the final pressure of 3lb. a pressure of but 1lb. is left to overcome the friction of the engine; not a sufficient quantity if the view that the terminal pressure must, to produce good results, be enough to overcome engine friction as well as back pressure. We are, however, inclined to think that this view is not altogether correct. The period during which the pressure is below that necessary to balance back pressure and engine friction is very small, and it is not quite so certain that a loss accompanies expanding to pressures below that balance. It is difficult to see where the line could be drawn if terminal

pressure is to be anything more than that necessary to balance back pressure. We could not make it back pressure and friction, or we might next have to take in friction of belts and shafting. We should therefore, from our point of view, say that Fig. 41 is a good diagram representing good practice.

It will be observed that the expansion line of the diagram between the division lines 1 and 2 is a good deal above the so-called isothermal expansion curve shown in dotted lines, and that from its intersection of the line 3 it follows as near as possible the isothermal or hyperbolic curve. In this respect it will be seen to differ from the diagram Fig. 39, p. 64, which is from a non-condensing engine, but working with the same range of expansion, namely, ten. It is not an uncommon thing to find the diagrams from condensing and non-condensing engines differing in this respect, and it is not easy to say why it should be so. The greater part of the expansion line of the simple engine diagram Fig. 39 is several lbs. above the hyperbolic curve; while Fig. 41, from the condensing engine, only shows an excess of actual over theoretic pressure during about one-tenth of the stroke at the commencement of the expansion. In both cases the most usual explanation of the excess pressure would be found in an appeal to re-evaporation at a low pressure of steam condensed at a high pressure during admission. We shall return to this subject.

It is, we assume, unnecessary to give any further examples of diagrams, merely as pressure diagrams, or of the use of the mean pressure obtained from them in calculating the indicated horse-power, so that we will in this case content ourselves with giving the diagram, with the pressures marked off. The diagram was taken with a spring which gave 16lb. to the inch, a scale which is suitable for the speed, but a spring which would not be stiff enough for higher speed.

It will be observed that this diagram (like Fig. 40) is accompanied by a series of diagonal and horizontal lines. These we may at once say are lines which have been drawn as the means of obtaining graphically the theoretic expansion curve, and for obtaining the point of cut off when that is not

known, or for assigning a point which shall give that at which cut off would have to take place in order to produce the curve of the indicator diagram, supposing no steam to be undergoing expansion except that accounted for by the admission part of the stroke of the piston. There is, however, besides this, an amount which is considerable admitted at every stroke to fill up clearance and port spaces, and this amount appears in the after part of the diagram, and gives a curve which appears to indicate a later cut off than is actually the case. This brings us to a consideration of the theoretical diagram of expansion.



PART IV.

EXPANSION OF STEAM AND THE THEORETIC EXPANSION CURVES.

IN engines of good design and in good order the steam pressure during expansion follows within narrow limits the curve drawn in accordance with Boyle's law, namely, that the pressure falls as the volume increases. The actual or indicator diagram curve falls more quickly at the commencement of the expansion than the law requires, but more slowly than it requires during the latter part of the expansion. This happens not to be so in Figs. 39 and 41 as regards the first part of the expansion, but it is borne out by Fig. 39 as regards the latter part of the stroke, and will be shown more completely by diagrams to be given hereafter. Practically, the departure by a minus quantity at the beginning, and by an excess in the latter part of the expansion, makes the indicator diagram curve nearly identical in area with the theoretical or hyperbolic curve formed according to Boyle's law, and this is partly due to the steam admitted to fill clearance and port spaces, and partly to condensation in the beginning of the stroke. We shall therefore assume for purposes of calculation that the expansion of steam in a steam engine cylinder takes place according to this law, and that the curve representing the diminishing pressures due to increase of volume is a portion of a hyperbola. To the accidental reasons for the approach practically to the theoretic diagram, according to this law, although theoretically there are reasons why it should not, we

shall refer hereafter, and will now proceed to the construction of the hyperbolic curve with a given base line. This we may do by calculation, or it may be done graphically, and as the former will afford some explanation of other matter as we proceed with it, we will take that first. Let

L = length of stroke in feet.

l = portion of the stroke during which steam is admitted, or the point of cut off, in feet.

s = any greater part of the stroke measured from the commencement in feet.

P = total initial pressure, *i.e.*, pressure in lbs. per square inch, including atmosphere.

P' = total pressure in lbs. per square inch at end of a given part, s , of the stroke.

P'' = total final or terminal pressure in lbs.

L , l , and s may, of course, be taken in inches, but for convenience we will take them in feet, as given above, because the stroke being expressed in feet and pressure in lbs. we get foot-pounds without further calculation.

Then the pressure at any given part of the stroke or

$$P' = \frac{P \times l}{s} \text{ or } P' = \frac{P'' \times L}{s}, \text{ and } P'' = \frac{P l}{L}.$$

We have several times referred to the spaces not swept through by the piston, including the clearance between cylinder cover and piston and the steam ports. When dealing with a practical case the volume of this clearance must be reckoned in such terms as will enable it to be added to the period of admission of steam before cut off; for instance, if a cylinder be 100 in. area, then each inch of the stroke of the piston will represent 100 cubic inches of steam, and cut off at one-fourth of the stroke, if the stroke be 20 in., will represent 500 cubic inches, *i.e.*, with cut off at 5 in. of the stroke. But, if the clearance between the end of the cylinder cover and the piston and the space in the ports, which it must be remembered has to be filled, be, say, 50 cubic inches, then the amount of steam

admitted will be equal to the 500 cubic inches plus the 50 cubic inches, which makes the total quantity admitted equal to that which would enter during 5.5 in. of the stroke, and this will of course affect the curve of expansion, although the 50 cubic inches of steam do no work during admission.

In the following example, however, we will assume that there is no clearance, and will return to an allowance for this hereafter.

In Fig. 42 let the base line $m n$ be the length of the stroke, say 6 ft., $m c$ the initial pressure, say 63 lb., $c d$ the period of admission, say one-third of the stroke. Draw the perpendicular $d a'$ from the point of cut off, and divide the period of expansion $d' n$ into any suitable number of parts. We will divide it

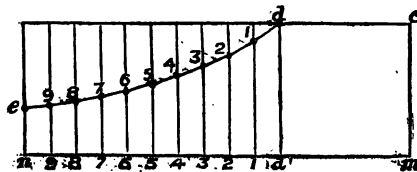


FIG. 42.

into ten parts, not as we have done in the case of the practical diagrams already given, namely, divided the whole length, including admission $m d'$, into that number. We will then calculate by the first of the foregoing expressions $P' = \frac{P l}{s}$, the

pressures at each of these divisions, 1, 2, 3, &c., and set them off by a scale of pressures on the vertical ordinates. The curve $d e$ traced through the ends of the ordinates is the hyperbolic curve of expansion. At the successive points of the base of the expansion line, each of the divisions being $\frac{1}{10}$ ths of a foot, the expansion part of the stroke being 4 ft., the pressures so found will be as follows:—

$$\text{At 1 it will be } \frac{P \times l}{s} = \frac{63 \times 2\text{ft.}}{2.4\text{ft.}} = 52.5\text{lb.}$$

$$\text{At 2 it will be } \frac{P \times l}{s} = \frac{63 \times 2\text{ft.}}{2.8\text{ft.}} = 45.0\text{lb.},$$

and so on for the whole of the ten pressure ordinates taken, or pressures at d' , &c.,

d' , 1, 2, 3, 4, 5, 6, 7, 8, 9, n
are ... 63, 52.5, 45, 39.4, 35, 31.5, 28.6, 26.1, 24.2, 22.5, 21lb.

These pressures being set off on their respective lines, and a curve drawn through the points so marked off, we get the hyperbolic curve $d e$.

The hyperbolic curve, which, as already stated, is the form of curve obtained by plotting ordinates of pressure and

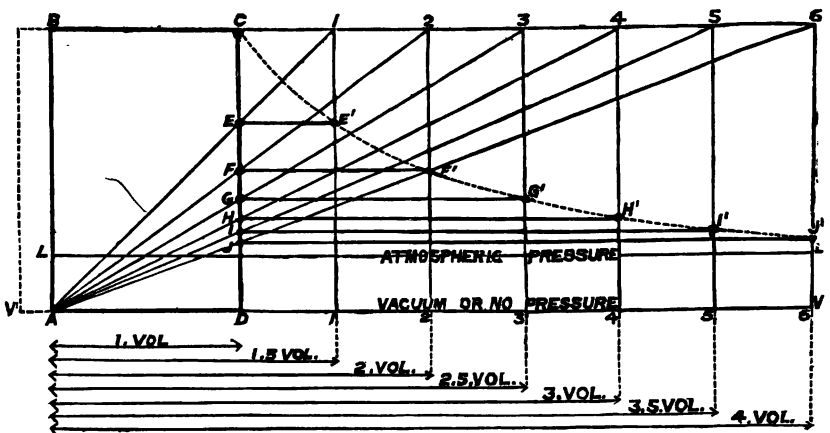


FIG. 43.

abscissæ of volume of a successively expanded gas, may be constructed by several graphic methods, one of which we will now take as being the simplest for indicator diagram work, as the lines drawn to divide the diagram into a number of equal parts readily form construction lines for this purpose.

In the diagram (Fig. 43) let LL be a line representing the atmospheric pressure by its distance above the line V'V, on a scale of 50lb. to the inch; the line V'V will then represent the line of true vacuum or of no pressure, and will be 14.70lb. per square inch below LL. The line V'V, as usual,

also represents volume, and the vertical ordinates represent pressure. $V'V$ may also represent the stroke, or that part of the cylinder through which the piston moves. The length AD is that part of the stroke throughout which steam is admitted to the cylinder. Upon AD erect the verticals AB and DC , their height being such as to represent, say, 75lb., atmospheric pressure included. At a scale of 50lb. to the inch the height AB will then be 1.5in. The rectangle $ABCD$ will then represent the volume and pressure of the steam during admission, or may represent one volume of a gas. We will assume that admission takes place during one-fourth of the stroke; the remainder of the stroke we divide into any number of parts, which may be equal parts. In our diagram each part is taken as half AD . We will now suppose the one volume is allowed to expand into a space equal to one volume and half, or to expand to line 1, 1. Now, drawing the line $A1$, we cut the line CD at E , and by producing E parallel to $V'V$, we cut the line 1, 1 at E' , and this is the second point in our hyperbolic expansion curve, C being the first point. Similarly, by drawing the lines $A2 A3 A4 A5 A6$, we cut the line CD at points $F G H I J$, and these points produced cut the lines 2 3 4 5 6 at points at $F' G' H' I' J'$, which are the succeeding points in our hyperbolic curve. It will be seen that when the original volume has been expanded into two volumes the pressure is halved; when expanded to three volumes the pressure is reduced to one-third, as will be found by measurement. The lines below the diagram show this at a glance. We adopted this method of constructing the hyperbolic curve in diagrams Figs. 39 and 41. By reference to these it will be seen that in Fig. 39, which is an indicator diagram from a high pressure non-condensing engine, the pressure throughout the greater part of the stroke is maintained much above that assigned by the law of relation between volume and pressure, the hyperbolic curve being much below the actual curve, showing that a very large amount of steam is condensed in the early part of the stroke and re-evaporated during the latter part, or that the steam ports and clearance spaces were very large, so that

a great volume of steam was necessary to fill these, this volume coming into play during the expansion part of the stroke and maintaining the pressure. In the diagram it will be seen that we had added about 4 per cent. for the volume of the steam filling these spaces, but it was probably very much more than this, 7 per cent. being a common quantity.

The properties of the hyperbolic curve enable us to assign a point in the stroke to which steam must be admitted to give the indicator curve of expansion, or rather a point to which steam would have to be admitted to give that curve if steam were not condensed in the early part of the stroke in unusual quantities. This

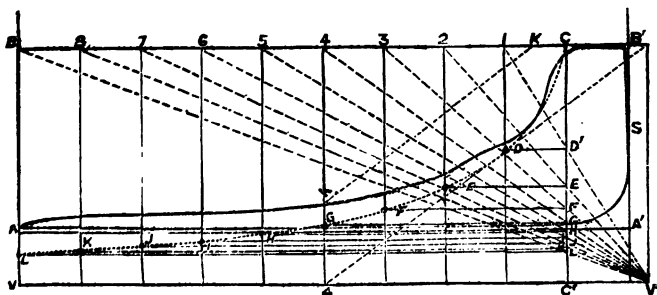


FIG. 39. (This figure is repeated for the convenience of the reader.)

point is given at K on the line BB' of Figs. 39 and 41, and is found as follows:—A point in the indicator diagram is chosen at which the expansion curve has become steady, as at K on the line 4, 4, Fig. 39, or on the line N5 on Fig. 41. The line 4, 4 or N5 being extended to the absolute vacuum line, the line 4 B', Fig. 39, or NN, Fig. 41, is drawn, and parallel thereto is drawn the line KK, starting from the point chosen in the indicator diagram curve. The point K on the admission line represents the point at which steam would have to be cut off to give the curve actually obtained, were it not that initial condensation causes the admission of a quantity of steam which is afterwards re-evaporated at the lower pressures during expansion. The distance along the steam admission line at the top of the diagrams, NK (Fig. 41), or B'K (Fig. 39),

represents, then, the volume of steam necessary to produce the curve the indicator has given.

Turning again to Fig. 41, it will be seen that although the range of expansion was very large, the actual indicator diagram curve follows very nearly the theoretical curve drawn by allowing about 7 per cent. for steam used in filling ports and clearance spaces. The actual curve is rather higher at the commencement, and only a little lower at the latter end of the

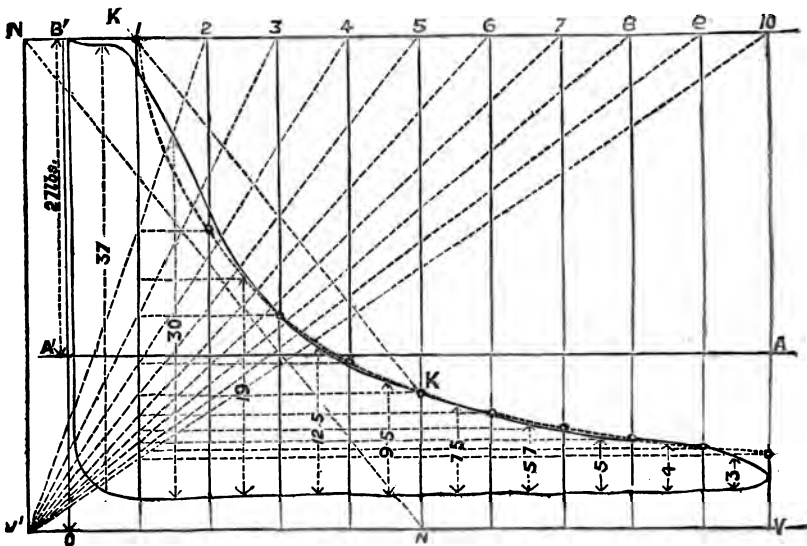


FIG. 41. (This figure is repeated for the convenience of the reader.)

stroke than theory demands. The engine from which the diagram Fig. 41 was taken was a condensing engine working with a good steam jacket, while Fig. 39 was taken from a cylinder without a jacket, and from an engine using much higher initial pressure.

In further illustration of Boyle's law, we may take the simple diagram given at Fig. 44. Here A D E C represents the original volume. Now, suppose the volume doubled and

the pressure correspondingly reduced to one half. We then get the rectangle, whose height is $D'E'$ instead of DE , and E and E' form two points in a hyperbolic curve. By again doubling the volume, we get a rectangle whose height is $D''E''$, and the corner E'' of the rectangle forms the third point in the curve. By again doubling the volume, so that it fills a space AD'' , we get a rectangle whose height is $D'''B''$, and the corner B'' forms the fourth point in the curve. By again doubling the volume the fifth point E and the curve may be indefinitely extended in this direction. On the other hand, if, instead of expanding from DE , we compress, say, double the pressure and half the

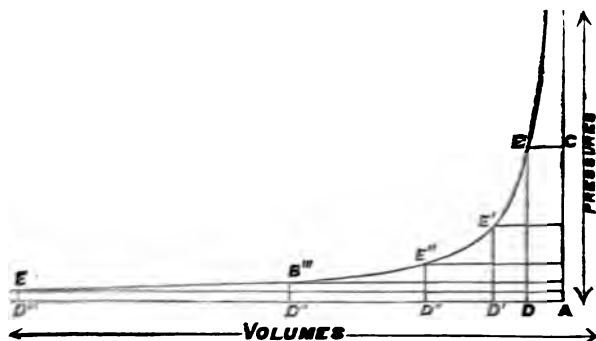


FIG. 44.

volume, we get a point by which the curve may be indefinitely extended upward to represent compression instead of expansion.

The hyperbolic curve of expansion is often called the *isothermal* curve of expansion, because a gas such as air gives this curve when it expands from one volume to a larger volume, providing that the thermal conditions remain the same. The pressure of such a gas varies inversely as the space it occupies only so long as the temperature remains constant. Thus a curve representing the fall of pressure of a gas as inversely proportional to its volume represents that pressure, when, and only when, the expansion takes place in a

vessel in which the air is maintained throughout the whole period of expansion at the same temperature, or when it takes place *isothermally*. Now, it only happens, that under ordinary conditions of working in well made steam engine cylinders, the curve of pressures of the expanding steam very nearly follows this hyperbolic curve or so-called isothermal curve. It must be remembered that, with respect to the expansion of steam, it is not an isothermal curve, inasmuch as the temperature of steam in any cylinder falls during expansion. Thus, with steam at 75lb. absolute, *i.e.*, including the pressure of the atmosphere, expanded down to a pressure of 5lb. absolute, the temperature falls from 307·5deg. to 162·4deg. The hyperbolic curve does not, therefore, represent isothermal expansion of steam, as that cannot take place; it only represents the isothermal expansion of a gas and approximately represents the relative pressures and volumes and correspondingly varying temperature of expanding steam.

As a means of obtaining the mean pressure throughout the stroke when a diagram cannot be taken, or for estimating the probable indicated horse-power of an engine of given dimensions, steam pressure, and cut off, a formula based on Boyle's law, and containing a hyperbolic logarithm, is used, and is as follows :—

$$p = P \times \frac{1 + H}{R};$$

p being the mean pressure;

P the initial pressure;

R the range of expansion,

H the hyperbolic logarithm of R

Thus for a pressure of 75lb. absolute cut-off at one-fourth the stroke, or in other words used with a range or ratio of expansion of 4, for which the hyperbolic logarithm is 1·386, we have $p = 75 \times \frac{2·386}{4} = 44·7$ lb. per square inch as the mean pressure.

In the steam engine cylinder, if of a condensing engine, this will be lessened by the imperfect vacuum, or by the back

pressure. This will vary in different engines, but is seldom less than from 3lb. to 4lb.

The mean pressure thus obtained is the same as that given by a hyperbolic curve, and it must be remembered that the steam admitted to fill the clearance and port spaces must be taken into consideration. The power of the engine during the steam admission period is not affected by this amount of steam, but the total mean pressure throughout the stroke is increased by it. Thus, in the above case, when the clearance steam is taken into account, the ratio of expansion is lessened, while the period of steam admission and the stroke remains the same, a condition not easily taken into the calculation, though very easily measured on a diagram. It is clear that the pressure is maintained at a higher pressure than it would be during expansion were it not for the steam so admitted, and we can only take it into the calculation by allowing for its volume, and assuming a smaller range of expansion. For instance, in the above case, if the clearance represented a volume equal to one-eighth the capacity of the cylinder, then, instead of the ratio of expansion being $\frac{1}{4}$, or $\frac{2}{3}$, it would be $\frac{3}{8}$, or R would be 3, but the work done would not be in proportion to this, as the full pressure steam would not have effect on the piston during more than the $\frac{1}{4}$ stroke.

The difficulty is that by adding the amount of the clearance to the steam admitted to the cylinder before cut-off we charge the piston with a greater duration of full pressure than it actually has, for the steam admitted to fill clearance spaces does no work during admission, yet we must credit it with doing work during expansion. The only possible way of calculating the mean pressure throughout the entire stroke is thus to take the whole of the steam admitted, and after obtaining the mean pressure from this, to deduct the amount of work that would be done by the steam admitted to fill clearance, if it were admitted during the movement of the piston instead of before it. Thus, taking the above case, the mean pressure, the hyperbolic logarithm 3 being 1.098, will be $75 \times \frac{1 + 1.098}{3} = 52.4$. This is the pressure during the whole

length of the cylinder, including clearance as part of the cylinder, assumed as above to be nine units of length. But the piston is not able to make any use of the pressure existing in the first ninth of the cylinder, this part or ninth being the clearance. This amount must, then, be credited to the piston. It may be thus done as a comparative example, taking the number of parts into which we have divided the stroke and the cylinder, including clearance, as multiples of length :—

| | |
|--|-----------------------------|
| Total pressure throughout cylinder and | |
| on unit area | $= 52.4 \times 9 = 471.6$ |
| Less ineffective pressure during clearance | |
| period or part of admission | $= 75.0 \times 1 = 75.0$ |
| Total pressure throughout stroke | $= 396.6$ |
| Mean pressure throughout stroke of piston | $= \frac{396.6}{8} = 49.57$ |

This is not absolutely accurate, but it is sufficiently accurate for all practical purposes. It shows how very important a part clearance may play, not only in explaining the much higher pressure which is found in cylinders during expansion than would be obtained were it not for the steam used in filling clearance spaces, but it shows how large a quantity of steam may be lost by clearances when sufficient compression is not employed, or when it cannot be employed. It will also be seen that the effect of clearance is greater with high pressures and early cut off or long ranges of expansion than with low pressures and small ranges of expansion. We need not follow this with reference to adiabatic expansion, as this, though of importance in theoretical investigations, is not so in practice.

If steam be expanded in a non-conducting cylinder, and neither receives nor loses heat by conduction or radiation, it is said to be expanded adiabatically. In this case the pressure falls in proportion to the amount of work done by the piston. If the work done is 772 foot-pounds, then the thermal equivalent of this, namely, one unit of heat, is abstracted from the steam, and the pressure falls in proportion to that one unit. At least, this is assumed

to be the case, and although there is no direct experimental proof that the performance of work of itself causes the abstraction of heat, it is reasonably assumed that this is the case; and as a large quantity of heat is lost, it is assumed that this is one of the sources of loss. In as much, however, as non-conducting and non-radiating cylinders do not exist, the curve which represents adiabatic expansion is not of much importance in this connection. We are not, however, clear that adiabatic compression or an approach to it does not occur when considerable compression is employed. Under adiabatic conditions the pressure, according to Rankine, varies nearly as the reciprocal of the tenth power of the ninth root

of the space occupied, or P being pressure, $P \propto \frac{1}{v^{\frac{10}{9}}}$. To find

the pressures from Rankine's formula involves a good deal of calculation, but a table of constants is given by Mr. Northcott in his work on the steam engine, which facilitates the calculation, and from this we may take a few figures (Table A.),

obtained by the formula $\frac{p}{P} = \frac{10 - 9r^{\frac{1}{9}}}{r}$, p being the mean pressure.

TABLE A. ADIABATIC EXPANSION.—*Constants for Mean Pressure.*

| Ratio of Ex-
pansion | Con-
stant. | Ratio of Ex-
pansion. | Con-
stant. | Ratio of Ex-
pansion. | Con-
stant. | Ratio of Ex-
pansion. | Con-
stant. |
|-------------------------|----------------|--------------------------|----------------|--------------------------|----------------|--------------------------|----------------|
| 1.5 | 0.931 | 3.5 | 0.622 | 5.5 | 0.467 | 7.5 | 0.373 |
| 2.0 | 0.833 | 4.0 | 0.571 | 6.0 | 0.438 | 8.0 | 0.357 |
| 2.5 | 0.748 | 4.5 | 0.529 | 6.5 | 0.414 | 8.5 | 0.341 |
| 3.0 | 0.678 | 5.0 | 0.495 | 7.0 | 0.393 | 9.0 | 0.327 |

To find the mean pressure exerted through the stroke when expansion takes place under these conditions it is only necessary to multiply the constant opposite the ratio of expansion. Thus, dry steam at 75lb. per square inch expanded adiabatically from one volume to four volumes will give a mean pressure

of $75 \times 0.571 = 42.82\text{lb.}$ The loss of pressure as compared with steam expanded in a jacketed cylinder is thus $44.7 - 42.8 = 1.9\text{lb.}$

An adiabatic curve of expansion may be constructed by taking out any number of final or terminal pressures—that is to say, pressures at any number of given ranges of expansion, as follows. The terminal pressure with adiabatic expression will be

$$p' = \frac{P}{R^{\frac{10}{9}}}$$

For example, we may take the following diagram, Fig. 45, which is the same as Fig. 42, page 75, but with the dotted curve added, showing the curve of expansion when adiabatically effected. In this diagram ABCD represents the unit

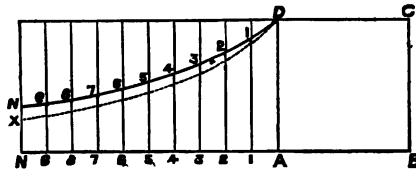


FIG. 45.

volume admitted before cut off. Estimating the pressure at the line 2-2, that is after expansion has increased the volume from 1 vol. to 1.4 vol., each of the spaces on this diagram being $= 0.2$, or two-tenths of the volume represented by distance AB, and taking the initial pressure as 63lbs. as before, we have $p' = \frac{P}{R^{\frac{10}{9}}} = \frac{63}{1.4^{\frac{10}{9}}} = \frac{63}{1.45} = 43.4 \text{ lbs.}$

In this way we may get the pressures at any number of points.

Taking our diagram they are on the ten pressure or pressures at D, &c. :—

| | | | | | | | | | | |
|-----|-------|-------|--------|-------|-------|-------|-------|-------|-------|-------|
| D, | 1, | 2, | 3, | 4, | 5, | 6, | 7, | 8, | 9, | N. |
| 63, | 51.6, | 43.4, | 37.38, | 32.8, | 29.1, | 26.2, | 23.8, | 21.8, | 20.2, | 18.5. |

The several pressures being set off as ordinates we get the adiabatic curve D, X, which is below the hyperbolic curve D N by the amount which the steam pressure would lose if it were expanded in a perfectly non-conducting cylinder.

The following (Table B) gives terminal pressure ratios for a few ranges of adiabatic expansion, or $R^{\frac{10}{9}}$.

TABLE B. ADIABATIC EXPANSION.—*Terminal Pressures.*

| Ratio of
Expansion R. | Adiabatic
Ratio $R^{\frac{10}{9}}$. | Ratio of
Expansion R. | Adiabatic
Ratio $R^{\frac{10}{9}}$. |
|--------------------------|---|--------------------------|---|
| 1.1 | 1.11 | 5.0 | 5.98 |
| 1.2 | 1.22 | 5.5 | 6.64 |
| 1.3 | 1.34 | 6.0 | 7.32 |
| 1.4 | 1.45 | 6.5 | 8.00 |
| 1.5 | 1.57 | 7.0 | 8.69 |
| 2.0 | 2.16 | 7.5 | 9.38 |
| 2.5 | 2.76 | 8.0 | 10.08 |
| 3.0 | 3.39 | 8.5 | 10.78 |
| 3.5 | 4.02 | 9.0 | 11.50 |
| 4.0 | 4.66 | 9.5 | 12.22 |
| 4.5 | 5.32 | 10.0 | 12.94 |

Our means of comparing an indicator diagram with the diagram which would be obtained from a steam engine were it not for practical sources of loss more or less unavoidable, but which steam engine builders do their best to make as small as possible, would be incomplete without the curve which is obtained by plotting the pressures and volumes of a given weight of steam at different temperatures. We have thus far only dealt directly with pressures and corresponding volumes of steam, temperature having been taken into consideration indirectly only, though we have referred to the condensation of steam during the commencement of the stroke of the piston consequent upon the entrance of the steam to a cylinder

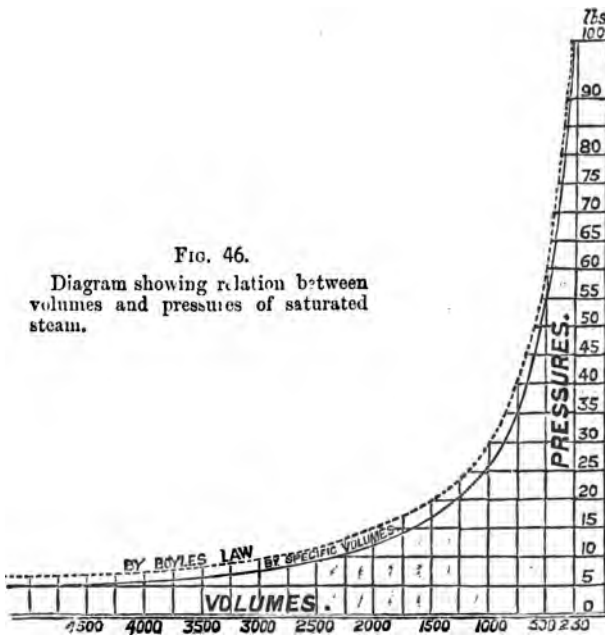
which has been cooled down to the temperature of the exhaust, or approximately thereto, and to the consequent fall in pressure. We have also referred to the rise in pressure of the steam during the expansion part of the stroke, consequent upon the re-evaporation during expansion of the steam so condensed, that re-evaporation taking place as the pressure falls to that under which the condensed steam will boil, and continuing to do so until exhaust takes place. It would be more correct to speak of this as a maintenance of pressure rather than a rise of pressure, as the re-evaporation does not cause a rise but retards the fall of pressure. It must be remembered that the pressure and temperature of saturated steam—that is, steam in the presence of the water from which it is evaporated—rise together, though not exactly in the same ratio. Every given weight of saturated steam has a definite volume, pressure, and temperature. That is to say, any given weight of water will give a specific volume of steam at a specific temperature or pressure. This is known as specific or relative volume. By plotting specific volumes and pressures we obtain a curve which differs somewhat from the so-called isothermal curve or hyperbolic curve, and from the adiabatic curve, and is in many cases rather nearer the actual indicator diagram than either of these. It is, however, almost as often so, partly as a result of a combination of contributing causes, including clearance, initial condensation, and subsequent re-evaporation, which in other cases may operate to make it incorrect, but inherently it is more correct as a theoretical curve than either of the others. It deals with steam as steam, and represents graphically the results of the experiments of Regnault, Fairbairn, and others, made under conditions which are very much those of the steam engine, except as to the sources of loss above mentioned. The following table gives a few corresponding temperatures, pressures, and volumes merely for convenience in dealing with the curve to which we refer, such a curve being very useful as a comparison with diagrams obtained from the steam engine cylinder.

TABLE C.—Relative Pressure, Temperature, and Volumes of Saturated Steam.

| Pressure in lbs.
per sq. inch. | Temperature.
Fahr. | Specific volume.
Water at 40°,
being 1. |
|-----------------------------------|-----------------------|---|
| 5.00 | 162.37 | 4627 |
| 10.00 | 193.29 | 2429 |
| 14.68 (atmospheric) | 212.00 | 1702 |
| 15.00 | 213.07 | 1669 |
| 20.00 | 228.00 | 1280 |
| 25.00 | 240.20 | 1042 |
| 30.00 | 250.40 | 881 |
| 35.00 | 259.3 | 764 |
| 40.00 | 267.3 | 676 |
| 45.00 | 274.4 | 608 |
| 50.00 | 281.0 | 552 |
| 55.00 | 287.1 | 506 |
| 60.00 | 292.7 | 467 |
| 65.00 | 298.0 | 434 |
| 70.00 | 302.9 | 406 |
| 75.00 | 307.5 | 381 |
| 80.00 | 312.0 | 359 |
| 90.00 | 320.2 | 323 |
| 100.00 | 327.8 | 293 |

The figures of the first and last of these columns set off on the vertical and horizontal scales, as in the accompanying engraving (Fig. 46), give us the curve of specific volumes, from which it will be seen that the volumes do not quite bear an inverse ratio to the pressure; and, in fact, the departure is considerable, the specific volume curve leaving the dotted curve, traced according to Boyle's law, by a perceptible distance even at the higher limits. This really is the curve we should obtain from the steam engine if the cylinder were at the temperature of the incoming steam, and fell in temperature only in accord with the fall in pressure, due to increase in volume during expansion. In constructing this curve, Fig. 46, the vertical line of pressures is divided off into equal parts to a suitable scale, and the student would find it useful to construct the curve on a large scale, say quarter of an inch to 1lb. pressure.

The horizontal base line is also divided off to a similar scale, each division representing, say, 50 volumes. In Fig. 46 each division represents vertically 5lbs. and horizontally 250 volumes, and it will be seen that for working purposes a much larger scale is desirable for minute accuracy in the curve. Along the line representing 5lbs. mark a point at a distance equal to 4,627 volumes, and so on, marking the specific volume



off on the lines of each of the pressures. It is usually most convenient, however, to commence at the upper part of the curve. Thus, having determined that the curve shall commence at 100lbs. pressure, the first vertical line erected upon the horizontal base may represent a volume of 250, as in Fig. 46. Now, on the horizontal line drawn at 100lbs. pressure, the volume at that pressure, as given at Table C, page 88, namely 293, must be marked, and will form the first point in the

curve. On the 90lb. line the volume 323 must be marked, thus forming the second point in the curve. On all the other pressure lines points must similarly be marked representing the volumes belonging to those pressures, the last and lowest on the curve, Fig. 46, being the 5lb. line, the corresponding volume being 4,627. Now, by drawing a line connecting all the points so laid out, the curve of specific volumes is obtained, as in Fig. 46. At a pressure of 55lbs. the curve falls almost exactly upon the corner of one of the squares made by the intersections of the vertical lines erected upon the base line of volumes, and the horizontal lines projected from the vertical line of pressures. By drawing a large curve with smaller pressures and volumes, represented by the intervening spaces between the vertical and horizontal lines, the specific volume at any pressure is much more readily and accurately seen, as will now be understood. The space between each vertical line in Fig. 46 representing 250 volumes, a volume of 30 is scarcely detected. This is seen by reference to the volume shown for 20lbs., which appears to be very nearly 1,250; but on turning to the Table C the true volume is found to be 1,280.



PART V.

COMPOUND ENGINES AND COMPOUND ENGINE DIAGRAMS.

WE may now turn our attention to the compound engine, and should, perhaps, first explain that the engine at present made under this name is old in conception, has been made for many years, but has become a more simple and more economical engine by modern developments. For these developments we have been chiefly indebted to marine engineers, who have by their practice shown that not only can high pressure be used with an approximately equable turning moment on the crank-shaft, but that the reduction in size of engines obtained by this use of high pressure is attended with no disadvantages, and the engines are more economical than the simple engine. The reduction in size of the engines and boilers which has become possible owing to the use of the high pressures and several-stage expansion is, on the other hand, of very great advantage on board ship, where every cubic foot of space gained in this way represents space acquired for cargo, and space which earns money on every voyage made. The compound system has now been developed to the extent of performing the expansion in three cylinders, and more recently in four cylinders, or by what are known as triple and quadruple expansion compound engines, and almost all new vessels are being fitted with them. The adoption of the compound system has extended to land engines of almost every

kind, and most recently to locomotive engines; and it is especially suitable for electric lighting work, because this work is uniform, and the advantages of the compound engine can only be fully obtained when the work it performs is approximately uniform.

We have already referred to the condensation which takes place in steam engine cylinders as a result of the fall in temperature of the cylinder, due to the fall in pressure during expansion. It is obvious that the condensation due to this cause will be greater as greater ranges of expansion are adopted. The limit to useful expansion in a single cylinder is thus controlled, and therefore the limit to useful high-pressure is also soon reached. For instance, in the case of a simple engine no good purpose can be served by adopting a terminal pressure of less than about three to five lbs. per square inch. On the other hand, the advantages of high ratios of expansion are counterbalanced by the condensation which attends them. The higher limit of pressure will thus be fixed by selecting the most economical range of expansion. If this be, say, six, then allowing for condensation, which is inevitable even at this ratio, and taking into consideration the economical advantage which attends the use of higher pressures in making engines smaller for the same work, and thus taking less steam for moving their own parts, the maximum useful pressure will not exceed from 40 to 50lb. per square inch, except where high pressure is specially useful in reducing the size of the engine.

A comparison of the possible advantages may be better or more readily seen by taking pressures and ratios of expansion, and, for the sake of illustration, making no allowance for losses. Thus a sixfold expansion in a simple engine would, if the terminal pressure be 5lb., need an initial pressure of 30lb. The specific volume of steam at 30lb. (see p. 88), is 881. If now, on the other hand, practical conditions are so altered that we can safely and economically use any range of expansion, the terminal being as before 5lb., and the initial pressure 150lbs. (quite common now), we could use a range of $150 \div 5 = 30$. Now, the specific volume of steam at 150lb. per

square inch being 203, one volume of such steam expanded 30 times would give us 6,090 volumes. On the other hand, the 30lb. steam would give $881 \times 6 = 5,286$. In practice, however, such a very large ratio of expansion would be impossible, the losses attending the use of the very high-pressure and temperature would be considerable, and five cylinders would be required to perform the expansion even in five-fold stages. With 150lb. steam, a ratio of from 15 to 20 might be used, and with the 30lb. steam not more than 3; so that the economical advantage is still, in spite of practical obstacles, on the side of high-pressure, while the engine for the purpose will weigh less and occupy much less space, a gain which is much greater when the space occupied by the boilers is taken into consideration.

The compound engine, therefore, whether with two, three, or four cylinders in series, is gaining ground rapidly, and for such purposes as electric lighting, driving mill machinery, or as marine engines, will no doubt wholly replace the simple engine. We must, then, consider the application of the indicator to engines of these classes, and then deal with the problems which appear in connection with the several diagrams from the two or more cylinders. In the compound engine with two cylinders in series, the steam from the boiler is first admitted to a smaller cylinder, where the cut-off will be from one-third to, say, one-half the stroke, or the ratio of expansion from 2 to 3. From this cylinder the steam is exhausted, or is allowed to pass into a larger cylinder either with a receiver between the two, or with only the steam valve chest and passages between the two cylinders. The steam is perhaps cut off at from half-stroke to three-fourths in this cylinder, according to its size. The relative sizes of the cylinders and the cut-off in both cylinders are arranged so as to obtain as nearly as possible the same amount of power from both. The pressure in the low-pressure or second and larger cylinder is always active as back pressure in the high-pressure cylinder, and thus cutting off in the low-pressure cylinder only tends to increase the back-pressure in the high-pressure cylinder towards the end of its stroke.

This may not be harmful, but the amount of this will be determined by the same considerations as those which would determine the most useful amount of compression in a simple engine, and by the amount that may be usefully used when necessary to balance the power of the two cylinders by this means. Assuming the two cylinders to be proportioned exactly to the pressure and work to be done, then the total expansion in a two-cylinder compound engine will be equal to the volume of the low-pressure cylinder divided by the volume of the steam admitted to the high-pressure cylinder to the point of cut-off, and in this case the back-pressure in the high-pressure cylinder will not increase towards the end of the stroke of the low-pressure cylinder, as the cut-off in the low-pressure cylinder will be practically at the end of its stroke

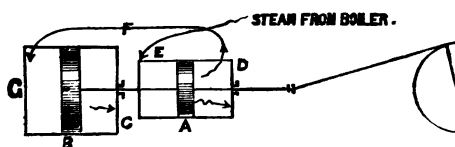


FIG. 47.

Compound engines are variously arranged with respect to the relative positions of the cylinders. The most usual arrangements are those having two cylinders, one behind the other, tandem fashion, with one piston rod running through both pistons; and those in which the two cylinders are placed side by side, and have their pistons connected, through the medium of connecting rods, to cranks at right angles, or nearly so. A third arrangement consists of two cylinders placed side by side, the pistons always moving in opposite directions through their connection with cranks at opposite or nearly opposite centres. There are other arrangements of the two-cylinder compound engine; but they are not now in such extensive use. The three arrangements above mentioned are typically shown by the diagrams, Figs. 47, 48, and 49. The tandem engine is shown by Fig. 47. In this A is the high-pressure cylinder,

and B the low pressure. Steam alternately enters at either end of the cylinder A, and is cut off at some part of the stroke of its piston, as at half stroke. Near the end of its stroke exhaust commences, the exhaust steam passing into the cylinder B. In the diagram (Fig. 47), steam from the boiler is entering at the back end of the high-pressure cylinder at E, and

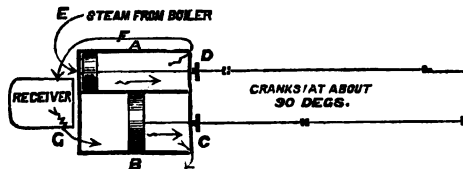


FIG. 48.

exhaust from the front end D of the same cylinder is passing, as shown by the arrow F, into the back end of the low-pressure cylinder B. An expansion valve must be used for the high-pressure cylinder, but need not be for the low. The pistons are shown as at half stroke, and the crank is therefore not quite vertical. The rates and direction of movement of the two pistons in this arrangement are, of

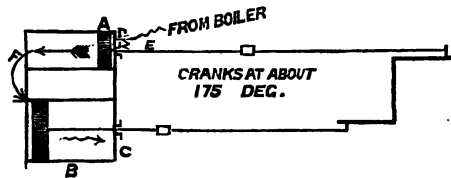


FIG. 49.

course, the same ; and while the advantages connected with the performance of the expansion in two cylinders are obtained, it is a disadvantage, from a mechanical point of view, that the maximum pressure is on both pistons at the same time in the same direction, and therefore also on the crank. The turning moment is subject to the same extremes as with a single cylinder engine, the maximum load on the two pistons

acting precisely as though the whole work were done by one piston. It will be observed that the back pressure on the high pressure piston is always the same as the forward pressure on the low-pressure piston, with the exception of the slight loss due to friction in the steam ports or passages. It may further be observed that the temperature in the high-pressure cylinder will have nearly as great a range as with the single cylinder, or simple expansive engine; but the range is spread over two strokes instead of one. If we follow the steam through one revolution, by the aid of Fig. 47, this will be seen. As indicated in the diagram the out stroke is being made as shown by the arrows placed inside both the cylinders. The steam passing from D to G will fall in pressure in accordance with the increase in volume, and the temperature with it, and as the high-pressure cylinder is in communication with the low-pressure throughout the whole of one stroke—that shown as being made—the temperature of the communicating vapour may fall to that of the low-pressure cylinder at the end of its stroke. The cylinder walls are prevented from falling to this temperature by the high-pressure steam the other side of the piston, but, of course, at the expense of the heat of that steam. Thus, with the tandem compound engine without a receiver and cut off valve, the effect of communication with the low-pressure cylinder when the steam has reached its minimum temperature is not avoided, and one of the chief objects of the compound system is only partly realised. On the other hand, the back pressure on the high-pressure piston falls to that of the lowest pressure in the low-pressure cylinder.

In the arrangement shown by Fig. 48 the range of temperature in the high pressure is not so great, and is never greater than that due to the difference between the maximum temperature in the two cylinders. This is effected by causing the exhaust from the high-pressure cylinder to pass into a receiver, as shown by the arrow from D. The temperature of the high-pressure cylinder will thus be not less at any time than that in the receiver. The back pressure in A is, also, never less than that in the receiver. The low-pressure cylinder B

receives its steam from the receiver, and is fitted with an expansion valve, or a valve with some lap, by which steam is cut off at less than full stroke, and communication between the two cylinders closed. The difference between the possible efficiency of this form of engine and that shown in Fig. 47 is not necessarily very much, but the gain is decidedly in favour of the receiver arrangement shown in Fig. 48. Sometimes no special receiver is made, the communicating passage and low-pressure valve chest being of sufficient capacity for the purpose. As will be seen from the diagram, the pistons are connected to cranks, which are placed on the two arms of a right angle, or at about 90 degrees apart, so that when one piston is at the end of its stroke the other is nearly at mid-stroke. The position of the cranks has some effect on the form of the indicator diagram, and as will be seen hereafter, an expansion valve is necessary for both cylinders. The action of the engine is the same as though it consisted of two distinct engines, the one supplied direct from the boiler, and the other from the receiver, the latter acting to a considerable extent in making the back pressure in the small cylinder more regular than in the arrangement shown in Fig. 47. The low-pressure cylinder may exhaust into a condenser, the engine then acting as though it comprised a high-pressure non-condensing engine and a low-pressure condensing engine.

In Fig. 49 another arrangement of compound engine is shown, in which the cylinders are placed side by side, in some cases with a receiver between, and in some—as is also the case with the arrangement shown by Fig. 48—with no more receiver than is provided by the passage from the high to the low-pressure cylinder and the valve chest of the latter. The cranks are sometimes placed opposite each other or at 180 deg. apart; but in other engines the cranks are placed a few degrees less than this apart, say at 170 deg., so that the engine may be started from any position, the efficient action of the steam not being at all affected. The admission and exhaust of the steam from the cylinders are the same as in the tandem engine (Fig. 47), and, although the pistons travel in

opposite directions, the rotative effort on the crank is the same, and the indicator diagram shows the action of the steam to be the same.

These diagrams (Figs. 47, 48, and 49) are sufficient to explain the principle of the compound engine, although other arrangements have been made, but are not general. We may, however, describe some by reference to Figs. 50, 51, and 52.

Fig. 50 shows an arrangement which is classed amongst compound engines, but is a single-cylinder engine in which the expansion is performed in two stages. A hollow piston rod or trunk piston rod is employed, and the connecting rod pivoted therein, as shown. The trunk occupies a large space, and thus reduces the area of the cylinder at the front end to a determinate amount less than that of the back end, and thus makes it suitable for the high-pressure steam. This steam

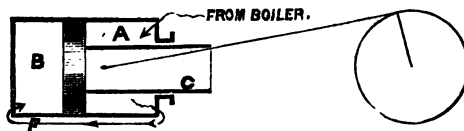


FIG. 50.

from the boiler is admitted to A, and being cut off at, say, half stroke, is correspondingly expanded. At the end of the stroke the steam passes as indicated by the arrow F to the end B of the cylinder, which not being occupied by the trunk is much larger in area, and serves as a low-pressure cylinder in which the steam already partly expanded, is expanded to the lowest pressure desired. The action of the steam is the same as that in the arrangement shown at Fig. 49, and the indicator diagram the same. It is an arrangement in which the temperature in the cylinder varies between the same extreme as in a simple engine, but, as in Fig. 47, the variation is spread over two strokes instead of one, and to this extent the arrangement is supposed to secure a possible economy not thought attainable by the simple engine. The use

of the trunk is not, however, advisable, as it not only provides a considerable condensing medium in consequence of its being half its time out of the cylinder, but the friction of the large gland necessary to make it tight is heavy.

Fig. 51 shows an arrangement in which the steam from the

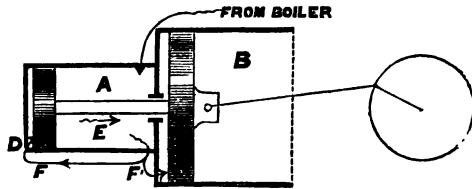


FIG. 51.

boiler enters the front end of the high-pressure cylinder, as shown by the arrow. When the stroke is completed the steam is exhausted simultaneously from the front end of the cylinder A into the back ends of both A and B cylinders, as shown by the arrows F and F'. The high-pressure piston thus performs the stroke, in the direction of the arrow E, in equilibrium, the pressure on both sides of it and on the low-pressure piston being the same, the low-pressure piston

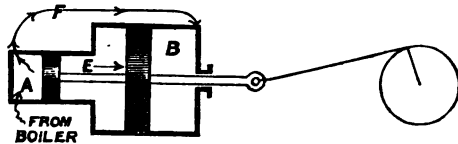


FIG. 52.

performing the work by virtue of its greater area. The front face of the piston in B is open to the atmosphere. It will thus be seen that the high-pressure steam from the boiler does not work against any back pressure, but the low-pressure piston has the atmosphere against it. If the end of the cylinder B were closed, as indicated by the dotted line, the tandem engine of Fig. 47 would be repeated.

Another form of engine is shown at Fig. 52. In this the

steam from the boiler enters at A, drives the piston in the direction shown by the arrow E, and when it arrives at the end of the stroke, the steam is exhausted or passes from A to B, as shown by the arrow F. In the arrangements shown at Figs. 50 and 51 only one slide valve, double ported, is required, and it is possible to do with only one for Fig 52.

It will be readily understood that in the triple and quadruple compound engine the system of expansion in stages is the same as that described for double cylinder engines, the range of expansion and number of stages being merely increased. It is unnecessary to describe the engines, but we may now proceed to consider typical indicator diagrams from engines of the several arrangements described.

Having now sufficiently described the principle and mode of action of most of the systems of compound engine in use, we will, in the order in which we described them, deal with indicator diagrams taken from each kind; but we will first take an ideal diagram to show what we should expect to get from a compound engine if made of non-conducting materials, so that there would be no loss of heat by radiation or conduction, and also, supposing that there be no loss of steam in ports or spaces in passing from one cylinder to another. Then the diagrams (Fig. 53) would represent what the indicator would obtain from the two cylinders of an engine without an intermediate receiver. Steam enters the high-pressure cylinder, as shown by the line L B, and is admitted during about two thirds of the stroke, as seen by the line B C. From C to the end of the stroke the piston is moved by the expanding steam, as shown by the line C D. At D exhaust takes place, but instead of the steam passing into the atmosphere and the pressure falling to E, as it would in the simple engine, and be indicated by a line D E, it passes into the second cylinder, and the entrance line F G is drawn by the indicator pencil. As the second cylinder has not an inexhaustible supply of steam, but only that which has come from the high-pressure cylinder, the pressure falls from the moment the piston commences to move, and the whole admission line G H is an expansion line, and is seen, by

putting the two diagrams in the relative positions shown in Fig. 53, to be a continuation of the expansion line C D of the high-pressure cylinder.

Thus far we have looked only at one stroke of the high-pressure piston, namely, that from B to D, and the one stroke of the low-pressure piston from G to H. We have, however, to observe that whilst the low-pressure piston was performing its stroke with pressure varying as represented by the height above A L of the line G H, the high-pressure piston was performing its return stroke with a back pressure, D K, equal to the pressure in the low-pressure cylinder. The back pressure against the high-pressure piston is thus equal to that indicated by the shaded part of the diagram, and only

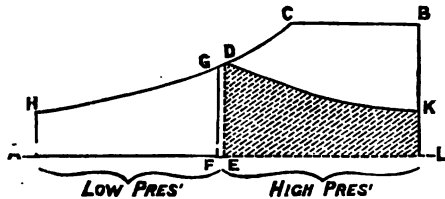


FIG. 53.

the area included within K B C D represents effective work. Such would be the diagrams under the ideal conditions. In practice, however, all the adverse conditions which we, for purposes of explanation, excepted, come into play, as will be more fully seen hereafter. We have, moreover, for simplicity of explanation, assumed that both the diagrams of Fig. 53 are taken with the same spring, or are of the same scale. This, of course, is not the case in practice, as the spring, which would be stiff enough to give a good diagram from the low-pressure cylinder, would be quite unsuitable and of insufficient strength for the indicator when on the high-pressure cylinder.

It is usual with engineers to compare the practical indicator diagrams from compound engines with the theoretical, and with each other, by reducing the length of the high-pressure

diagram in proportion to the relation between the area of the high-pressure piston and the low-pressure-piston. By this means the high-pressure diagram may be placed above the low-pressure diagram, as shown in Fig. 54, instead of end to end with both to one scale, as in Fig. 53, and not only is the expansion curve of the high and low-pressure diagrams brought into position for inspection as to the action and behaviour of the steam, but the amount of loss of steam pressure between the two cylinders, as shown by the gap which occurs between them even when they are placed as close as possible. This gap is seen

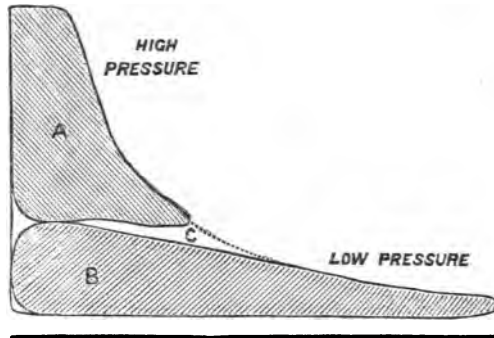


FIG. 54.

at C, Fig. 54, A and B being the high and low-pressure diagrams. These are, however, diagrams such as would be obtained from an engine with a receiver, while the two diagrams (Fig. 53) are supposed to relate to the theoretical engine without receiver and with cylinders assumed for explanatory purposes to be of the same size. Before following up the method pursued in redrawing actual diagrams for comparing them with each other and with theoretical expansion curves, we will look at a few actual diagrams.

Commencing with those from a small engine, we have the diagrams given in Fig. 55. These are from a small vertical tandem condensing engine. The cylinders were respectively

4.5in. and 7in. diameter, the stroke being 10 inches. The two diagrams were taken with the same spring, the pressure in the boiler at the time being 80lbs. to the inch. When both diagrams are taken with the one spring the low-pressure diagram appears small when compared with the high-pressure diagram, and the diagrams are not as correct as when a spring suitable to each pressure is employed. The engine made 180 revolutions per minute, and at this speed gave 12.53 indicated horse-power. An inspection of

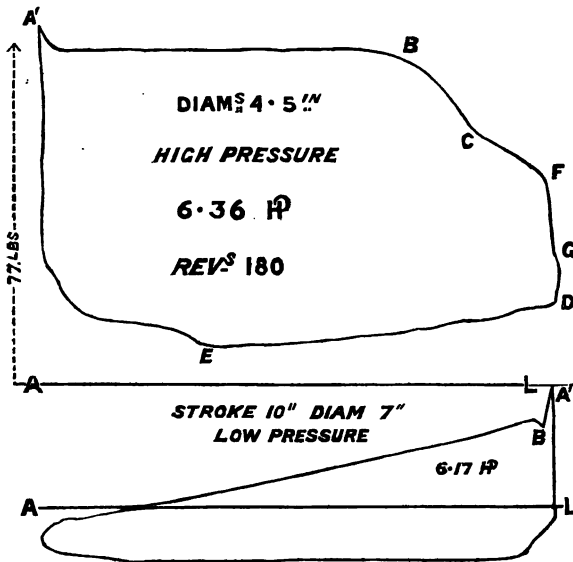


FIG. 55.

these diagrams reveals a very satisfactory division of the work between the two cylinders, but there is a good deal more loss of pressure by the steam in passing between the high and low-pressure cylinders than is necessary, and much of this seems to be due either to restricted passages or to unsatisfactory action of valves, though most of it was no doubt due to condensation in the exterior pipe con-

necting the low to the high-pressure cylinder and in the latter cylinder itself. It will be seen in both diagrams at A' that the momentum of the indicator parts under the influence of the pressure and speed carried the pencil above the actual pressure, the recoil carrying it down to B in the low-pressure diagram before steadiness was attained. Full steam, it will be seen, was admitted to the high-pressure cylinder during about three fourths of the stroke, cut-off taking place near B, from which point pressure fell very rapidly, but became suddenly checked in its fall at C, probably by re evaporation. Exhaust occurred freely at F, except for the slight check at G, probably caused by the low-pressure piston reaching the end of its stroke. From D the pressure falls steadily, though less fast than that in the low-pressure cylinder, until the point E is reached, the valve closes, and compression commences. The relations between the areas of the cylinders being as 15.9 to 38.48, and assuming the cut off to be at about three-fourths or 0.75 of the stroke in the high pressure, the actual range of expansion was only about $\frac{38.48}{15.9 \times 0.75}$ or 3.2. This is not enough, even in so small an engine, to secure the advantages derivable from the compound system. Such a range of expansion could be at least as economically used in a single cylinder or simple engine.

The diagrams (Fig. 56) are copied from a tracing of the indicator diagrams from a small horizontal tandem engine of the same class, namely, non-receiver class, but the passage between the two cylinders is proportionally much larger than in that from which diagrams (Fig. 55) were taken. It may be remarked that tracings from diagrams are not always exactly like the diagrams. This will be seen hereafter. The cylinders are 7.25 and 13.25 respectively, the stroke being 18 in., and the number of revolutions 100 per minute. The steam pressure in the boiler was at the time the diagrams were taken 65 lb., or an absolute pressure of 80 lb. The maximum pressure in the high-pressure cylinder reached 63 lb. or 78 absolute. Steam was cut off at about half stroke, and the mean pressure taken from the

two high-pressure diagrams is 39.3lb., and the corresponding mean in the low-pressure cylinder is 7.5lb., the pressures at either end of the low-pressure cylinder being more nearly alike than in the high-pressure, although even there the means are not remarkably dissimilar. It will be seen that the slow closing of the valves caused a gradual rise in the back pressure in the high-pressure, as shown by the line from E to B, or E to B'. The indicated horse-power as measured from these diagrams was 13.75 in the high-pressure cylinder, and 9.39 in the low, or a total of 23.14. For satis-

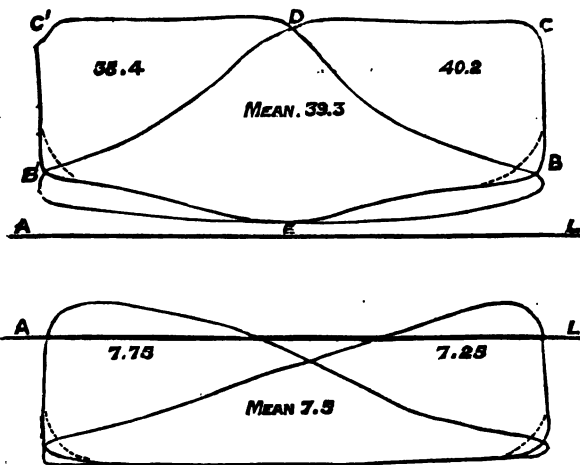


FIG. 56.

factory working the heel B B' of both diagrams should be less sharp. For 100 revolutions per minute, a curve such as shown by dotted lines would have given a better result with this engine. A slight difference in the rapidity of port opening is seen at C and C'.

Figs. 57 are from the same engine working at 90 revolutions per minute, and more cooling water passing through the condenser, thus producing a better vacuum. These diagrams have been more faithfully reproduced. They show better

action at the compression end of the stroke, but the improvement is marred by the defective action of the valves and by excess of compression, especially at B. Re-evaporation is plainly seen in the too slow and irregular fall of the high-pressure expansion line. An undesirable difference in the admission and early expansion of the right hand low-pressure diagram is seen at B, and it will be seen that this is accounted for by the shorter

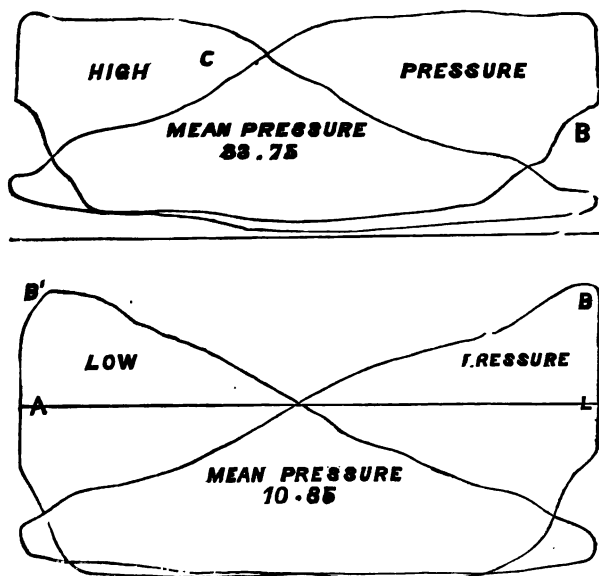


FIG. 57.

periods of admission to that end of the high-pressure cylinder from which the steam came, namely, the left-hand end, as seen at C. In both Figs. 56 and 57 the diagrams were taken to a scale of 30lb. and 8lb. to the inch respectively. We may now take an example from a large compound tandem with no other receiver than that provided by the large pipe connecting the two cylinders. The diagrams (Figs. 58 and 59) are the high and low-pressure diagrams from a large horizontal tandem

pumping engine, fitted with differential valve gear, and making from ten to eleven strokes per minute. They were taken to scales respectively of 32lb. and 16lb. to the inch, but, like the preceding diagrams, are reduced in size for our purposes. The engine cylinders are 45in. and 72in. respectively, and the stroke 9ft. The high-pressure diagrams, of which one pair is here shown, give a mean pressure of 40·75lbs. per square inch, and the area of the cylinders or pistons being 1,590 square inches, we have, with 10 strokes per minute

FIG. 58.

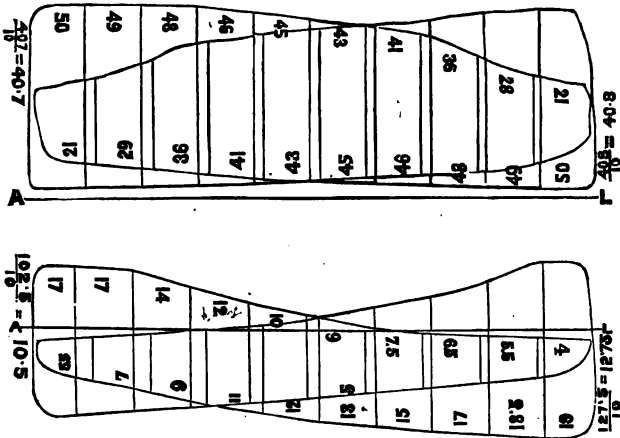


FIG. 59.

$(1,590 \text{ in.} \times 40.75 \text{ lbs.} \times 90 \text{ ft.}) \div 33,000 = 176$ horse-power indicated. Two corresponding low-pressure diagrams, taken immediately after, but at a time when the speed had reached 11 strokes per minute, gave a mean pressure of 11·5lbs., and the area of the cylinder being 4,071·51 square inches, we have $(4,071 \text{ in.} \times 11.5 \text{ lbs.} \times 99 \text{ ft.}) \div 33,000 = 140$ horse-power, or a total of 316 horse-power. It will be seen that these high-pressure diagrams are satisfactory in most particulars, except in the restricted admission shown by the right hand diagram;

but the low-pressure diagrams show an insufficient capacity in the condenser, the vacuum being but slowly formed, or, rather, very slow in reaching its maximum.

The diagram shown at Fig. 60 is taken from a tandem non-condensing engine of the type shown by Fig. 50. The engine, nominally of 10 horse-power, was tested in comparison with an ordinary 10-horse portable engine by the same makers, and was said to make a very satisfactory performance. Both high and low-pressure diagrams are taken by the same spring, and the arrangement of the engine gives the diagrams in a relative position which permits immediate comparison of the actual area with the total area it would have if there were no gap. That is to say the diagrams are placed with the exhaust end of the high-pressure diagram immediately above the admission end of

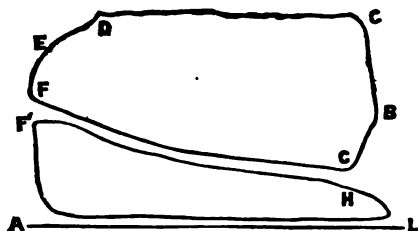


FIG. 60.

the low-pressure diagram. Following the steam, we may commence at B with the admission of steam from the boiler with the high-pressure cylinder, the pressure rising to C with but little restriction; full-pressure steam is admitted to D, expansion takes place to E, when exhaust takes place and pressure falls to F, the exhaust passing at pressure F into the low-pressure cylinder, wherein the pressure is shown by the line F' H, while the pressure in the high-pressure cylinder is shown by the line F G, and is higher than that in the low by the amount of the gap between them, the loss of pressure shown by the distance between the lines F G and F' H being caused by friction through ports and passages and valves. At G the valve closes the high-pressure

exhaust and compression takes place from G to B. In a trial with the engine of this type, as above alluded to, the compound engine made 18,000 revolutions against a given brake load, the water being maintained at the same level in the boiler as shown by the water gauge, while the non-compound engine made but 15,000 revolutions, and the boiler lost an inch in the level of its water, the same weight of coal being used in both cases. The boiler would thus have been able to continue this rate of work with the compound engine, while the simple engine would soon have had to be stopped, or have some of its load removed to enable the boiler to overtake its water consumption. Moreover, the compound engine boiler was fed with cold water, while the simple engine had a feed heater, which raised the water to 150 degrees.

In marine engines various modifications of the several types of compound engines which we have described have been made, and in many cases ordinary engines have been converted into compound engines by the addition of high-pressure cylinders, usually mounted above the original cylinders, thus making two tandem engines. In some cases one high-pressure cylinder has been used, the steam passing from it to both the existing cylinders of a pair of simple engines. In all cases, however, the object and the results are the same, namely, the use of high-pressure steam expanded in two stages instead of one, so as to obtain a smaller range of temperature in each cylinder than could be obtained with an equal expansion in one stage and one cylinder, and to secure the economy in steam which follows. More recently it has become the practice of shipowners to have their compound engines converted into triple-stage expansion engines, and this has been done in most cases by placing a small cylinder for the high pressure on the top of the original high-pressure cylinder of the compound engine. By this means a still higher pressure can be economically utilised without adding many new parts to the engine. New engines in this form are numerously made, but many engineers give preference to engines with three cranks, one for each cylinder. By the use of three cranks, the work done per

revolution of the crank shaft is done in three stages instead of two, and the stresses are correspondingly reduced, the reduction being in some cases as much as 50 per cent. We are not, however, concerned with the engines so much as with the diagrams from them, but it is necessary to be acquainted with all the types, or those mostly in use, as the form and reading of the indicator diagrams will both be affected by and depend upon the characteristics of the engines.



PART VI.

DIAGRAMS FROM COMPOUND ENGINES.

Of indicator diagrams from engines of the type shown by Fig. 48 we may now take some examples, and, firstly, a set from a semi-fixed compound engine of the type very largely used for driving small electric lighting installations. Figs. 61 and 62 are from such an engine doing about 50 indicated horse-power. The cylinders of the engine are 8.5 and 17.75

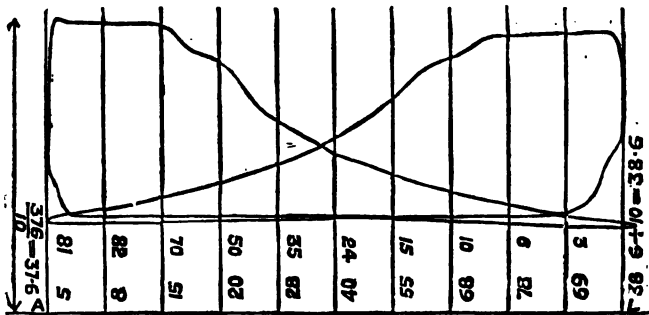


FIG. 61.

respectively, the strokes being alike. The cranks are at right angles, and there is a receiver of large size, in a position between the cylinders, and with them immediately below the smoke box, which retards loss of heat. Both cylinders are, moreover, jacketed with steam direct from the boiler.

From the high-pressure diagrams (Fig. 61) it will be seen that the steam from the boiler, at about 125lb. on the square

inch, is cut off at from one-fourth to one-fifth of the stroke, the expansion line being somewhat wavy, partly the effect of the oscillation of the indicator spring and parts, and partly due to re-evaporation, which is somewhat considerable, the pressures at any point after cut off being higher than can be accounted for by any other explanation, even when the amount of steam in clearance space is considered. Exhaust takes place at 40lb. per square inch, a pressure which is fairly uniformly maintained in the receiver. This back pressure is very large, but it not only secures a much lower range of temperature in the cylinder than is common in the high-pressure cylinder, and therefore less loss by cooling of the cylinder walls, but the maximum effective pressure on the piston, about 80lb., is low

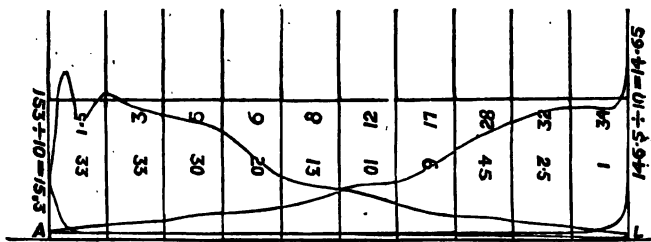


FIG. 62.

enough to make it unnecessary to use the very tight piston rings which by some makers are used with high pressures. The rings and piston springs are thus more easily kept in order. The loss of pressure between the high and low pressure cylinders is shown by these diagrams to be a little over 3lb. per square inch, which is small. Steam is cut off at about one-third the stroke, expansion taking place in the low-pressure cylinder, and much less than usual in the receiver, which thus acts very much as a boiler for that cylinder. When the diagrams were taken the engine was running at about 125 revolutions per minute, driving 40 arc lamps. The mean pressures in the high-pressure cylinder, as measured and marked on the diagrams, were 37.6lbs. and 38.6lbs., giving 24.8 and 25.4 indicated horse-power in the

front and back ends respectively, or a mean of 25.1 horse-power in this cylinder. The mean pressure in the low-pressure cylinder were 14.65lbs. and 15.3lbs., and 27.5 and 29.4 horse-power respectively in the front and back ends, or a mean of 28.4 horse-power. The amount of compression in either cylinder may be considered small considering the speed of the engine, but the results obtained as given by the makers, Messrs. Robey and Co., are highly economical. The mean of a number of experiments with the engine working up to 53 horse-power gave, it is stated, a water consumption of 986lbs. water and 99lbs. of coal per hour. These are exceedingly and almost phenomenally low figures for a non-condensing engine, the water consumption being but 18.6lbs. and the coal consumption only 1.86lbs. per indicated horse-power per hour.

We may now turn to a set of instructive diagrams derived, as explained, with reference to Fig. 48, from a set taken from an engine similar to that just described, and by the same makers, but of 40 nominal horse-power, instead of 20, and fitted with a condenser. Although taken with springs proportionate to the pressures in the cylinders, the two pairs of diagrams D, D' and L, L' have been plotted to the same vertical scale. In this form, however, they only serve as separate diagrams, showing the action of the valves, the cut-off, pressures, and the behaviour of the steam. In order to make them more correctly indicative of the performance of the engine and the steam it uses, we have reduced the horizontal scale of the high-pressure diagram by making it proportional to the relation between the areas of the high and low pressure pistons, so that the two diagrams are comparable directly, the scales being the same, and thus enabling us to see at a glance the difference between the diagram that theoretically would be obtainable from the two cylinders if acting as one non-conducting cylinder. The expansion curve CH is the curve AN reduced as described with reference to Fig. 54—that is to say, diagram D is reduced to diagram E, so as to be the same pressure and volume scale as diagram L. The continuation of that curve CH in dotted line HC is an

assumed continuation at the same rate of expansion. The space F represents loss of pressure, and its area will represent work lost by reduction of pressure by cooling and frictional influences, and the spaces between low-pressure steam chest and the low-pressure cylinder, *i.e.*, the port and clearance spaces. We have in these diagrams an opportunity of

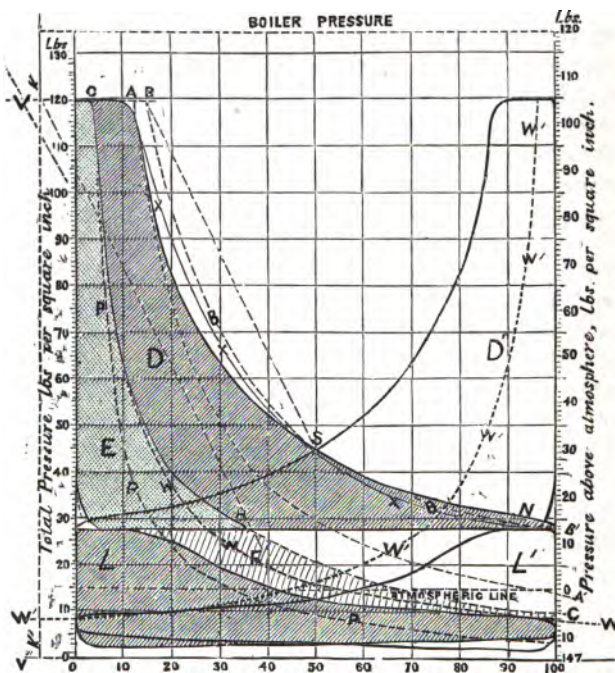


FIG. 63.

making use of the methods described with reference to Figs. 39 and 41, pages 78 and 79, of ascertaining the amount of clearance, and of showing graphically the difference between the theoretical and actual behaviour of the steam in the cylinders. We will take the full diagram D. Supposing the cut-off to have taken place at A on the line C A B, as the diagram D

shows, then, if the quantity of steam admitted to the cylinder were only as much as is indicated by the length of the admission line to A, the expansion line by Boyle's law would follow the dotted expansion curve A A'. The actual diagram, however, follows the line A N, which is very much above it, showing that a great quantity of steam in excess of that indicated by the length of the admission line, assuming cut-off to have taken place at A. As there is no doubt that cut-off did take place at that point, it is clear that the port and clearance spaces and initial condensation must together be assumed to account for the quantity of steam represented by the area between the curves A N and A A'. Adopting the method described in pages 78 to 81, we will assume the clearance and port spaces to equal 7 per cent. of the volume of the cylinder, and draw a line K' K to represent this volume added to that accounted for by the indicator. We may now select a point in the expansion curve whereat the expansion may be assumed to have become regular, as at S. By a vertical from this point to the line of no pressure we get the point 50, and may now get the diagonal 50 K, and parallel with it the line S B, the intersection of the admission line at B giving a length K B of admission line, which corresponds to such an expansion line as A N given by the indicator, but which would give us a true expansion line as shown by the dotted line B B B B'. As the point B is beyond the real point of cut-off A, it is obvious that more than 7 per cent. of clearance, port space, and initial condensation must be allowed for. We may find at once what this is. If we connect S and A by a line, and then draw one parallel to it from 50 we shall get the point V, showing the actual amount of steam admitted to have been 11 per cent. greater than is shown by the whole of the original indicator diagram D. Knowing now the real volume of steam concerned, we may by the means already explained construct the true or theoretic expansion curve for that volume and the pressures. This we have done, and it is shown by the fine line curve A X X B'. This it will be seen is, as usual, a little higher than the indicator curve in

the first part of the expansion, and below it in the latter part of the expansion or of the stroke.

Now, constructing the theoretic curve of expansion or isothermal curve for the diagram D, as reduced to the size of the part marked E, we get the curve marked C P P P, which is the theoretic expansion curve for the whole diagram, consisting of the combined high and low pressure diagrams D and L, the

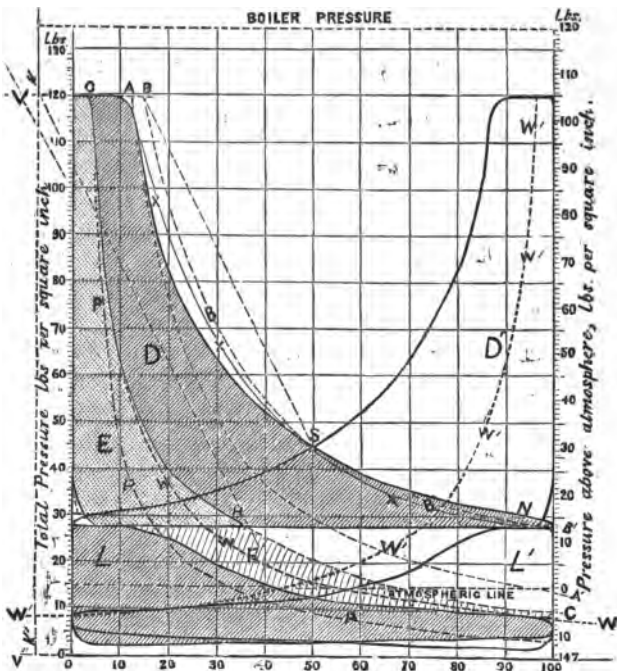


FIG. 63.

(This figure is repeated for the convenience of the reader.)

diagram D being reduced in scale to compare with L. This, however, does not take into consideration the 11 per cent. of clearance. Taking this in its proper proportion, remembering that the cylinders are as 3 to 1 in area, we get as the final and true expansion curve for the volume of steam

employed, the curve C W W W, which for clearness we have repeated on the unshaded diagrams D' and L', as shown by the curve W' W' W'. It will be seen that the theoretic curve is coincident with the curve CH from 120lb. down to 60lb. It will be further seen that the real gap between the expansion curve of the lower diagram and the curve W W is lessened by the shaded area comprised between the letters W W H. Practice and theory thus run very close together, and if we were to make a reduction in pressure corresponding to the heat converted into work, the gap shown would really be a very small one indeed.

It should have been mentioned that the dotted line at the heel of the high-pressure curve represents the compression reduced to the scale by which CH is produced.

The engine from which the originals of these diagrams was taken has cylinders 12 and 21 inches in diameter, the stroke is 2ft., the revolutions 90, and the horse-power indicated 90. It is briefly described in the *Proceedings* of the Institution of Mechanical Engineers for August, 1885.

It will be noticed that the highest pressure in the low-pressure diagram is shown the same as the exhaust in the high pressure. The upper part of the low-pressure diagram touches the lower part of the high-pressure diagram. This we must assume to be due to the influence of the heated receiver

We have said that, as shown by these diagrams, practice and theory run very close together. Perhaps it will be well to explain that by this it is meant that when the adverse conditions found in practice are all, as far as they are known, taken into consideration, then the expansion curve given by diagrams taken from a well-designed engine with cylinders and intermediate heater kept warm by heated gas in a smoke box, is very nearly the same as that which would be drawn from theoretical considerations. This, however, does not mean that practical results are equal to those which theory, apart from adverse practical conditions, would lead us to hope for.

In reducing the diagram D, Fig. 63, which is the same as that on the other side, marked D', to the size of the part shown by lighter shading at E, it will be observed that it is only the horizontal dimensions that are changed. The areas of the cylinders are, as already stated, as 1 to 3, and the strokes being the same, the volumes or the contents of the cylinders will, apart from clearance spaces, also be as 1 to 3. The high-pressure diagram is thus reduced in dimensions so as to be of the same volume scale as the low-pressure diagram, by making its horizontal dimensions, on any lines, one-third of those of the original diagram, assuming the original to be the same length as the low-pressure diagram. Thus, the length of the base of the diagram E, reduced from diagram D, should be one-third of the total length of the low-pressure diagram.

In further explanation concerning the indications of the diagram, Fig. 63, as showing the quantity of steam actually passed into the cylinder, it may be again remarked that the diagram D or the opposite one, D', does not indicate much more than 60 per cent. of the steam that actually passed into the cylinder, and that for the full quantity of steam that was used at each stroke the back or admission line of the diagram must be taken back to a vertical line, connecting V and V. Taking this line as the limit of the diagram reduced for combination with the low-pressure diagram L, we get, as explained, the hyperbolic curve C W W W as the curve of expansion of the steam actually admitted. We will return to this subject and apply the methods here explained by reference to diagrams, Figs. 64 and 65.

PART VII

COMBINED DIAGRAMS FROM COMPOUND ENGINES.

In proceeding further with our treatment of the diagrams from compound engines, we must take a practical example of the transfer of diagrams as taken by the indicator with different springs to corresponding diagrams of the same scale of pressures and volumes. This has been referred to with reference to Fig. 54, and at more length to Fig. 63. In the latter, the effect on the diagram of different and of the same clearance spaces was shown, with a method of ascertaining the clearance theoretically when the actual clearance was not known. One of the few recent complete tests of an ordinary small compound engine, of the type shown by Fig. 44, namely, that made with one of Messrs. Davey, Paxman and Co.'s engines, provides us with a set of diagrams in connection with which all the data are known. The trial was carried out by the late Mr. W. E. Rich, M. Inst. C.E., and Prof. Kennedy. The engine was one which had been driving dynamo-electric machinery at three successive Exhibitions held at South Kensington, London, namely, the Health, the Inventions, and the Indian and Colonial, and after all this work gave the excellent results which have been recorded. The principal particulars and dimensions of the engine are :—

| | |
|--|-----------|
| Diameter of high-pressure cylinder | 12·28in. |
| " low " | 20·08in. |
| Area of high-pressure cylinder..... | 118·25in. |
| " low " | 315·50in. |

| | |
|--|----------|
| Stroke of pistons | 2ft. |
| Revolutions per minute..... | 109·5 |
| Boiler pressure | 107lb. |
| Indicated horse-power | 109·5 |
| Nominal horse-power..... | 40·0 |
| Water consumed per ind. h.-p. per hour ... | 24·15lb. |
| Coal, Welsh " " " ... | 2·56lb. |

The engine was fitted with Paxman's automatic cut-off expansion gear. From this engine high and low-pressure diagrams were taken at regular intervals during the trial and simultaneously from both cylinders. Figs. 64 and 65 are engraved from one set.

There is some difference between the two diagrams from either end of both the high and low-pressure cylinders, but both are very good. The admission of the steam to the front end appears to be much better than at the back end, as, although a horizontal admission line is not obtained, the pressure reaches much nearer to that in the boiler. The closing of the valve took place somewhat slowly. The admission line at the back end is apparently better, but is not really any better, for the approach to a horizontal line is only obtained at the expense of a full admission in the earliest part of the stroke, the highest pressure at the front end being about 98lb., whilst that at the back end is about 10lb. less. Thus, unless the difference between boiler pressure and cylinder pressure is known a diagram which may really indicate a restricted admission, may appear better, or seem to indicate better performance, than one which has a descending admission line and belongs really to a better performance. Cut-off takes place in these diagrams at about half stroke, and the expansion follows approximately the hyperbolic, or what is generally spoken of as the isothermal expansion curve. The exhaust of the front end diagram will be seen to take place more nearly at the pressure in the small receiver—which is formed by the space between the cylinders—than in the back diagram, which would appear to be due to the admission of more steam to the back end than to the front end. The lower or exhaust

line of both diagrams shows the effect of the rise in pressure in the cylinder as the low-pressure piston approaches the end of its stroke, and thus causes rise in the receiver pressure, and consequently in the back pressure against the high-pressure piston. The amount of compression is small at either end.

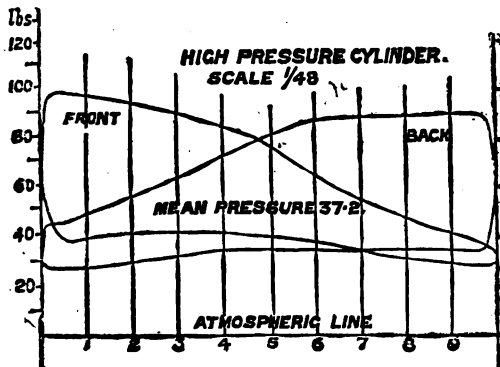


FIG. 64.

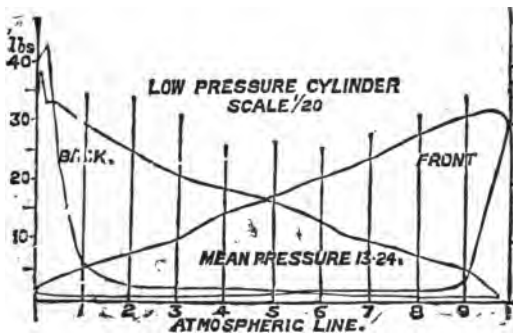


FIG. 65.

In the low-pressure cylinder (see diagrams, Fig. 65, taken with 20lb. spring) the amount of compression adopted is very considerable, and especially at the back end, where it carries the pressure considerably above that in the receiver,

from which it drops by the opening of the valve exactly at the end of the stroke, thus forming the loop shown. Judging by the back pressure shown by the low-pressure diagrams, although only about 2.5 lb., somewhat larger exhaust ports in the port face and valve would be advantageous. It is not, however, without diffidence that any such comment is made, for the efficiency of the engine is remarkably high, as shown by the figures which are given below the diagram, Fig. 66. The evident freedom of the exhaust at the commencement of the stroke may, perhaps, suggest a little hesitation on this point. It may, moreover, be remarked that, although it is supposed that the super-heating of the exhaust steam in the smoke box and chimney tend to reduce back pressure, it is not at all so certain that this is the case, although for other reasons economy may result from this position of the exhaust pipe.

We now come to the high and low-pressure diagrams, Fig. 66, as plotted from those we have been discussing, so as to give them the relative forms and areas they would have if taken from one cylinder, except for the gap which occurs through the performance of the expansion in two stages. The enlargement and combination in this manner affords a picture of the steam performance, which is instructive in many respects.

In setting out these combined or expanded diagrams, as they are often called, we first ascertain the cubic capacity of the two cylinders, including their clearance. In the case before us the volumes are 1.7 and 4.55 cubic feet, and of these the clearance represents about 0.08 and 0.2 cubic feet respectively, or about 4.5 per cent. and 6 per cent respectively, the diameters of the cylinders being 12.28 and 20.03, and their areas 118.3 and 315.5. We now draw a vertical line, EE' , and divide it into some convenient scale representing lbs. per square inch, say a quarter inch to the lb., carrying the scale sufficiently high to include the highest pressure occurring in the high-pressure cylinder, or, say, the boiler pressure. In our case we are dealing with a non-condensing engine, and therefore with pressure above that of the atmosphere, which will be our zero. For constructive purposes we must, however,

carry our scale to the vacuum line or absolute zero 14.70lb. below that of the atmosphere. At the atmospheric

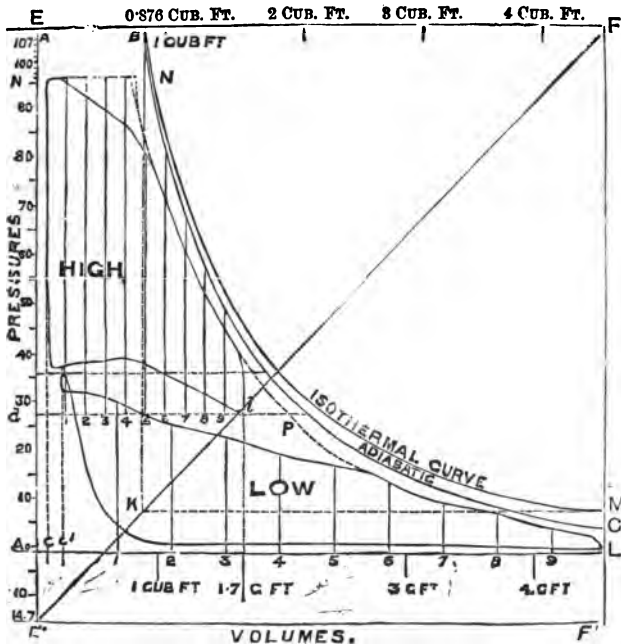


FIG. 66.

Diagrams from 40 H.-P. Semi-fixed Engine, by DAVEY, PAXMAN and Co., indicating 109·5 I.-H.-P.

| | |
|----------------------------------|-----------|
| High-pressure Cylinder | 12-28in. |
| Low-pressure Cylinder | 20-03in. |
| Stroke | 24 0 in. |
| Revolutions per minute | 104 |
| Pressure in Boiler | 107lbs. |
| Water per I.-H.-P. per hour..... | 24 15lbs. |
| Coal per I.-H.-P. per hour..... | 2 56lbs. |

Trial 27th October, 1886.

pressure line we draw the horizontal line A L. We may also draw the line E F at the level of the boiler-pressure, which

in this case is 107lb. We now divide the lines EF or AL by a scale which shall represent volumes, or the capacity of the cylinders in cubic feet; that of the low-pressure cylinder being 4.55 cubic feet we divide the lines into 4.55 parts. Now from the line AL near A we draw the light lines *c* and *c'*, at distances from the line EF which shall represent the volumes of the clearance spaces and ports, namely 0.08 and 0.2ft. We next draw a light line, *dl*, at the level of the lowest pressure in the high-pressure cylinder, as shown by the diagram in Fig. 64, taking that from the front end. On this line set off the length, equal 1.7 cubic feet, and from line C divide this into 10 parts. Similarly divide the line AL from line C' = 4.55ft. into 10 parts. From each of the 10 divisions on *dl* draw light vertical ordinates, and similarly from each of the divisions on the lines AL. Now, having divided the diagrams in Figs. 64 and 65 into 10 equal parts, we may transfer the pressures shown on each of these divisions to our new expanded diagram lines. For instance, on the line C we set off above AL the 97lb. found on the front end of Fig. 64, and also the lower pressure of 36lb. On the vertical ordinate 1 of this high-pressure diagram, Fig. 64, we see that the highest pressure was 95lb., and the lower 37lb. These we set off on ordinate 1 of the new high-pressure diagram. Similarly, we set off the pressures on the remaining ordinates 2, 3, 4, &c., until having 22 points at the upper and the lower ends of these pressure ordinates we have only to connect them to complete this diagram. In constructing the low-pressure expanded diagram, we take as the higher pressure on line C' the height of the loop at the commencement of the stroke, namely 37lb., and the lower part of the loop, namely 33lb., and on ordinate 1 we find the pressures shown were 30 and 7, and we set these off on the corresponding ordinate, on Fig. 62, and similarly the pressure on the other 9 ordinates, until the diagram is completed. We have now the two diagrams in their true proportions relative to those of the cylinders and the pressures in them, and can compare the behaviour of the steam in the cylinders with that which would theoretically be the behaviour in a non-con-

ducting cylinder. Knowing the point at which steam was cut off we can mark the volume of steam admitted to the high-pressure cylinder, plus that which is necessary to fill the clearance spaces on the line E F. The volume in this case is 0·876 cubic feet, as marked on the line E F at B. From this point we may set out the hyperbolic or so-called isothermal expansion curve B M in the manner explained with reference to Figs. 39, 31, and 43, or the adiabatic curve B C, as described with reference to Fig. 45. The area not filled by the diagrams between ALCNN may be said to be due to the losses which occur in the practical engine, chiefly due to condensation from different causes, only a comparatively small part being due to the performance of work. The largest area is that of the gap between the diagrams and within the dotted line P.

Since the date of the engine trial referred to in connection with Figs. 64, 65 and 66, the trials of simple and compound engines carried out at Newcastle in July, 1887, have been made, and the results show even greater economy. Nearly a dozen engines were tested at these trials, but the chief interest lay in those of three makers, namely, Messrs. Davey, Paxman and Co., Messrs. J. and H. McLaren and Messrs. Foden and Sons, all of whom entered both simple and compound engines. The highest results were, as will be seen from the annexed table, obtained by Messrs. Davey, Paxman and Co., both with their simple and their compound engine. All the engines were of the portable kind, those of Messrs. Foden and Sons being traction engines.

| Maker. | Lbs. Coal per Brake-H.-P. per hour. | | Difference. | Percentage of advantage in favour of compound system. |
|---------------------|-------------------------------------|-----------|-------------|---|
| | Simple. | Compound. | | |
| Davey, Paxman & Co. | 2 6 | 1·85 | ·75 | 23·9 |
| Foden | 2·76 | 1·94 | ·82 | 29·7 |
| McLaren | 2·68 | 2·18 | ·50 | 18·6 |

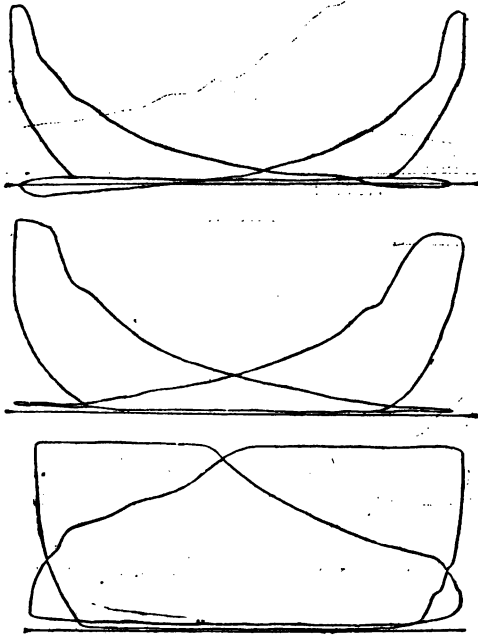
The results obtained by the simple engines of Messrs. Davey, Paxman and Co. and those by Messrs. McLaren show what a high pitch of perfection these engines have attained. The results are practically identical, and, even allowing for slight inaccuracy in the brakes, they show that a portable engine can be made now, which, with very careful stoking, will give a brake-horse-power with under two and three-quarters of a lb. of good Welsh steam coal. We shall refer to diagrams from these engines in succeeding chapters.



PART VIII.

DIAGRAMS FROM SIMPLE AND HIGH-SPEED ENGINES.

We may now turn our attention back to diagrams taken from different engines under similar circumstances, and from



FIGS. 67.

the same engine under different loads, and may commence with those from simple or non-compound engines.

Figs. 67 illustrate a set of diagrams taken from a recent engine fitted with Proell's automatic cut-off expansion gear,

as made by Messrs. Robey and Co. They were taken with the work varying from little more than that of engine and shafting friction, to that requiring steam to about five-eighths of the stroke. The Proell gear cuts off the steam instantaneously, whether cut-off occurs at one-twentieth or one-half the stroke, and the effect of the cut-off is thus immediately

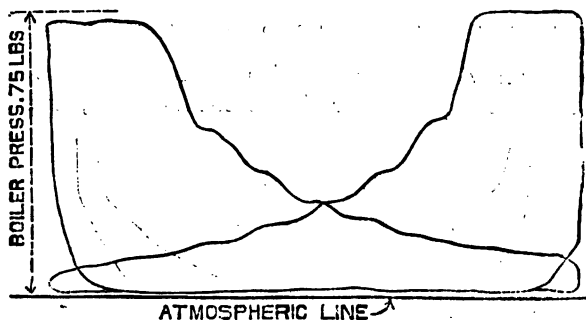


FIG. 68.

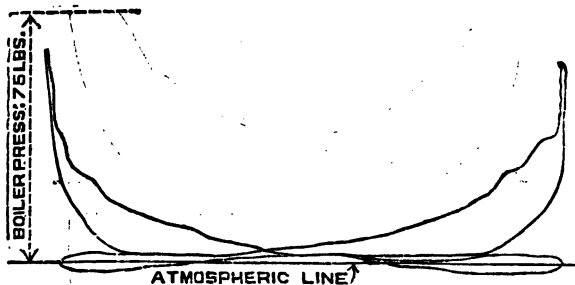


FIG. 69.

felt in the cylinder and shown in the diagram. This gear takes two forms—that in which it controls circular tappet valves, which themselves admit the steam to the cylinder, and that which controls one such valve, which admits steam to the steam chest, the steam being distributed by slide valves in the ordinary way. In the former the action of the cut-off gear is instantly felt in the cylinder, as shown by Figs. 67; but the action is not much less rapid with the second-mentioned form,

as is shown by the two diagrams, Figs. 68 and 69, taken respectively with steam cut off at about one-fourth stroke, and at the earliest fraction of the stroke which the gear permits. It will be seen that in all the diagrams taken at the very early cut-off the pressure during expansion falls below the atmospheric pressure, though not quite to the same extent; but it is curious that the fall is nearly as great with its cut-off at one-thirtieth as at about half this, an apparent contradiction due to the effects of condensation and re-evaporation, and to the fact that the steam in the clearance spaces constituted so large a portion of all that shown in the diagram (*see* Fig. 69). Upon



FIG. 70.

examination, it will be seen that steam was cut off immediately after the commencement of the stroke; but although this is the case, the diagrams show that there was a considerable quantity of steam ready to show itself as the pressure and temperature fell, most of which was in the port and clearance spaces.

As illustrations of very high efficiency in automatic expansion gear, Figs. 70 and 71 may be given; these are from an engine fitted with valves of the Corliss type, as made by Proell, and controlled by automatic gear of the kind which acts upon the excentric, and is a governor having weighted levers placed within the flywheel, or a wheel for the purpose.

The diagrams, Figs. 70 and 71, were taken with an engine running at 180 revolutions per minute, on full work, and very light. The irregularity of the expansion line in both diagrams is due to the indicator, which worked badly at this speed, or

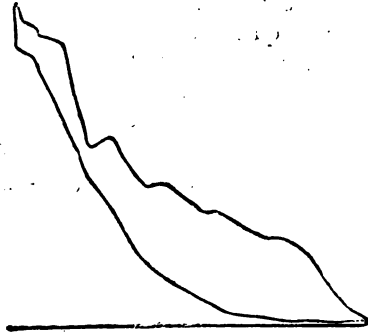


FIG. 71.

the diagrams would have been taken at a higher speed. They show, however, a very free admission for the steam when work has to be done; a sharp cut-off; a free and full exhaust, and a suitable compression, which is greater as the engine is



FIG. 72.

doing less work, as it should be, so as to fill the clearance spaces with steam of the previous stroke.

The diagram, Fig. 72, is from a similar engine with cylinders 14in. diameter, 20in. stroke, and running at 242 revolutions per minute, about 800ft. piston speed, driving the electric generators of the National Theatre at Buda-Pesth.

This diagram is excellent for so high a speed, though for high economy it might have been preferable had it expanded to a slightly lower pressure, and shown also a little more compression. It is, moreover, quite evident from the point of cut-off and the high pressures during expansion that there must have been large clearance spaces in this engine, and probably also large initial condensation, for the pressure during expansion is higher than it would be without these conditions and the cut-off shown.

In Fig. 73 are shown instructive diagrams from an engine which had originally a slide valve of short stroke, and, therefore, of slow and late cut-off, but which was afterwards fitted

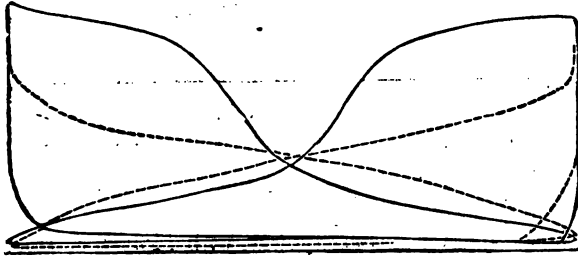


FIG. 73.

with an exhaust and expansion valve. The dotted line diagrams were taken when the old valve was in use. From these it will be seen that the valve opened the port so slowly that, although the full pressure was obtained in the cylinder at the instant the piston was at the end of the stroke, steam was admitted so slowly or rather through so small a passage that the pressure in the cylinder was not maintained. The port was, moreover, so slowly closed that steam was admitted through a great part of the stroke and wire-drawn nearly all the time. With this valve in use the engine worked very uneconomically. The one valve was removed and replaced by a ported valve of the simplest type, with Meyer expansion valves at its back. The result was the full line diagrams, and although the cut-off is not sharply defined its period can be

easily seen. With valves of a little greater travel the result would have been still better. As it was, the superior economy soon repaid the cost of the new valve gear and added very much to the power of the engine.

The diagrams, Figs. 74, 75, and 76, are from three American engines, and they show the effects of different ranges of expan-

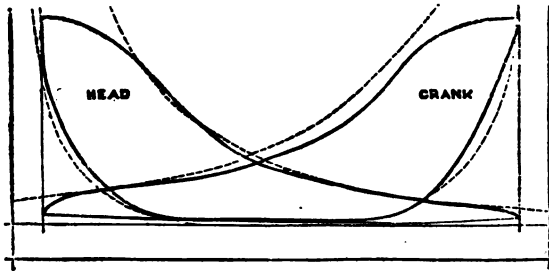


FIG. 74.

sion and different amounts of compression in high-speed engines. Fig. 74 is from a Porter-Allen engine, with a cylinder 11.5in. diameter, 20in. stroke, running at 227 revolutions per minute, and indicating 69.34 horse-power, with 44.3lbs. of steam per

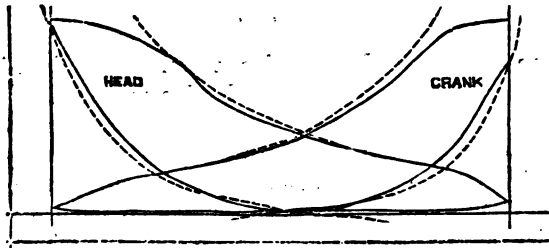


FIG. 75.

horse-power per hour, of which the indicator diagram showed 23.38lbs. In the diagrams it will be seen that the cut-off is not sharp at either end, that it is a little later at the crank end than at the head end, probably to allow for volume and cooling effect of piston-rod; that the actual diagram from the head end is very near the hyperbolic expansion curve, although

that at the crank falls below it from the commencement, and that the compression curve shows at both ends a more rapid rise in pressure than Boyle's law demands. This is probably due to the quantity of excess water present, which was evaporated during the latter part of the admission part of the stroke and during expansion. The rapid rise was also probably partly due to the heat generated in compression, the compression being excessive.

Fig. 75 is from a Southwark engine, with a cylinder 9in. in diameter, 10in. stroke, running at 305 revolutions per minute, and indicating 29.11 horse-power, with 46lbs. steam per horse-power per hour, of which the indicator accounted for 24lbs.

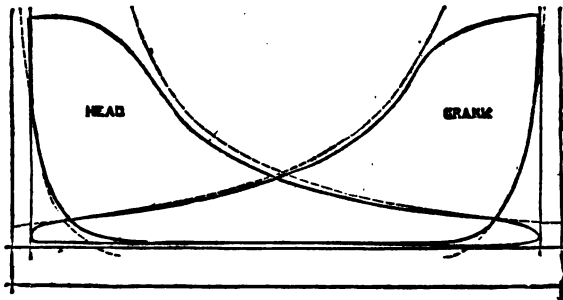


FIG. 76.

The main features of the diagram are much the same as those of the Porter-Allen engine, but with slightly less satisfactory cut-off at the back end, and with even more excess of compression at both ends. Both these engines are wasteful of steam, and use double that used by an English engine designed for the same work, but at a lower speed.

Turning now to Fig. 76, we have a better performance. The diagram is from a Buckeye engine, with a cylinder 10in. diameter, 20in. stroke, running at 201 revolutions per minute, and indicating 54.32 horse-power, with 30.93lbs., or say 31lbs. of steam per horse-power per hour, of which the indicator gave evidence of 20lbs. This engine is much the same size as the Porter-Allen engine, the diameter and the revolutions being a little less



and the power a little less, but it is far more economical than the others. From the diagram it will be seen that, although the cut-off is not very sharply defined, it is uniform, and that more steam is admitted than in the other engines, or rather the cut-off is later, which may be a different thing. The compression is, moreover, very much less and the speed is less. These diagrams and figures tell their own tale. They show that a range of expansion of more than about four or five in one cylinder is not economical; they show that, though moderate compression is essential to the satisfactory and economical working of an engine, too much may be wasteful, and they further show that at very high speeds an enormous quantity of steam is used in driving the engine itself, or through uneconomical conditions involved. Moreover, with a high speed, such as 300 revolutions per minute, the clearance spaces, or so much of them as is not filled by compression, have to be filled 600 times per minute as compared with, say, 300 for an engine running at a moderate speed, say, 150 revolutions per minute. It would, moreover, appear from these diagrams that large clearance space is accompanied by loss, although the space be filled by compression.

At pages 124-5 we referred to the performance of the engines tested by the Royal Agricultural Society in July, 1887. We may now return to these engines and give some diagrams from them. Messrs. Davey, Paxman and Co.'s simple engine had a cylinder 9.5in. diameter, a 12in. stroke, and it ran at 132 revolutions with a pressure of 105lb., and a brake load of 17 horse-power. The engine ran 4 hours 23 minutes actual and 4 hours 29½ minutes mechanical time, with a supply of 193lb. of coal, equivalent to 2.6lb. of coal per brake-horse-power per hour.

The simple engine of Messrs. McLaren had a cylinder 8.5in. diameter, a 15in. stroke, and it was run at 130 revolutions and with 135lb. pressure and a brake load of 17 horse-power, and made a run of 4 hours 23 minutes actual and 4 hours 34½ minutes mechanical time, with a coal supply of 199½lb., equal to a consumption of 2.68lb. per horse-power per hour.

1887

The simple engine of Messrs. Foden and Sons had a cylinder 7.5in. diameter and a 10in. stroke. It ran 4 hours 29½ minutes actual and 4 hours 30 minutes mechanical time, with a coal supply of 138½lb., equal to a consumption of 2.76lb. of coal per horse-power per hour. This engine was run at 168 revolutions and a pressure of 120lb., with a brake load of 12 horse-power.

The compound portable engine of Messrs. Davey, Paxman and Co. had cylinders 5.75in. and 9.25in. diameter, and a 14in. stroke. It was run at 134 revolutions, with a pressure of

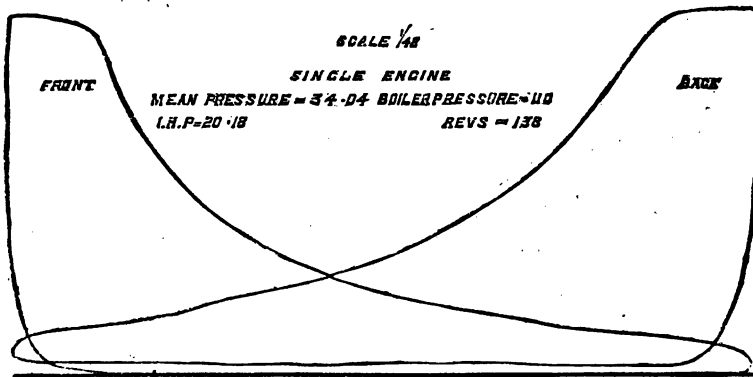


FIG. 77.

150lb., and a brake load of 20 horse-power. The engine ran 4 hours 28 minutes actual and 4 hours 39¾ minutes mechanical time, the coal used being 168lb., equal to a consumption of only 1.85lb. of coal per brake-horse-power per hour.

The compound engine of Messrs. Foden and Sons had cylinders 4.75in. and 9.5in., and 10in. stroke. It was run at the speed of 156 revolutions and a pressure of 250lb., with a brake load of 18 horse-power. This engine ran for 4 hours and 21 minutes actual and 4 hours 27½ minutes mechanical time, with a coal supply of 148½lb., equal to a consumption of 1.94lb. of coal per horse-power per hour.

The compound engine of Messrs. J. and H. McLaren had cylinders 5.75in. and 9in. diameter and a stroke of 15in. It was run at 135 revolutions and a pressure of 155lb., with a brake load of 20 horse-power. This engine ran for 4 hours 24 minutes actual and 4 hours 51½ minutes mechanical time, taking a supply of 202½lb. of coal, equal to a consumption of 2.18lb. of coal per brake-horse-power per hour.

These engines and the indicator diagrams from them may be taken as illustrative of the best modern practice in engines of the kind, and we therefore give, in Fig. 77, a pair of the diagrams from the simple engine for which the Royal Agricultural Society's prize was awarded.

These diagrams, it will be seen from what has now been said, are excellent. Rather more steam was admitted to the back end of the cylinder than to the front end, no doubt partly in greater clearance and port spaces at that end. Measurement will show that the curve at this end is higher than it could be with the quantity of steam necessary to fill the cylinder to the point of cut-off, much of this being no doubt due, as already explained, to initial condensation and clearance, especially as the compression was not sufficient to fill the clearance space at the full pressure of the incoming steam. A little more compression would probably have been of advantage, but with so remarkably high a result as was obtained with this portable non-condensing engine, it is not easy to speak with certainty of the probable result of any alteration.

With these examples, we may leave the simple engine and return to examples illustrative of practice with the compound engine.

PART IX.

DIAGRAMS FROM COMPOUND ENGINES.

We may in the first instance turn to some representing the earlier and still very much used form of compound engine, namely, the Woolf engine, chiefly used for pumping purposes.

Figs. 78 are diagrams taken from compound Woolf beam engines, as made for the West Middlesex Water Works by

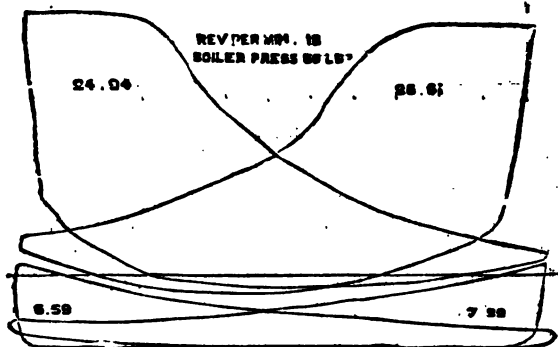


FIG. 78.

Messrs. Simpson and Co. These diagrams are from extremely economical engines, using only 14.56lbs. of feed water per indicated horse-power. The diagrams show a very satisfactory distribution of the steam. The actual power of the engines, calculated by the water lifted is 174, and the coal consumed

per actual horse-power per hour 1·826, and 1·54lb. per indicated horse-power.. The coal evaporated 9·56lbs. of feed water per lb. of coal from the temperature of the hot well, and, besides this, kept the jackets filled with steam. Thus the feed water consumed per indicated horse-power was only 14·56, but the feed water consumed was probably over 17lbs. per actual horse-power—a remarkably economical result.

Another good example of diagrams of this kind is given in Figs. 79, which show a pair reduced to the same scale taken from one of the compound Woolf beam pumping engines for

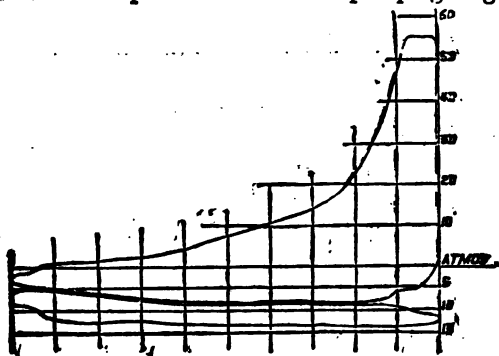
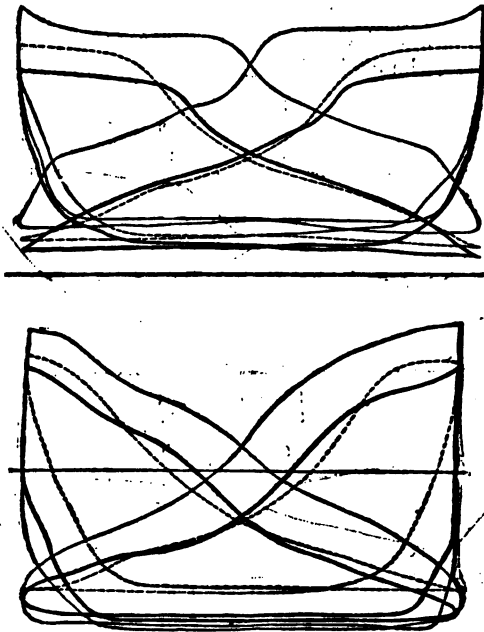


FIG. 79.

the Corporation sewerage works of Burton-on-Trent, made by Messrs. Gimson and Co., of Leicester. The engine has a high-pressure cylinder, 24in. diameter and 6ft. stroke, the low-pressure cylinder being 38in. diameter and 8ft. stroke. The ratio of volumes of the cylinders is thus, omitting clearance and port spaces, 1 to 3·5. The distribution of the steam is effected by tappet valves. It will be observed that, as in almost all these moderate-speed engines, the expansion gives a curve of the hyperbolic form, and is satisfactory in every respect, for, in copying from the original, the instep, which is shown on the high-pressure diagram, has been unintentionally exaggerated. We have not the steam consumption by these engines, but the coal consumption was 1·95lb. per indicated horse-power per hour.

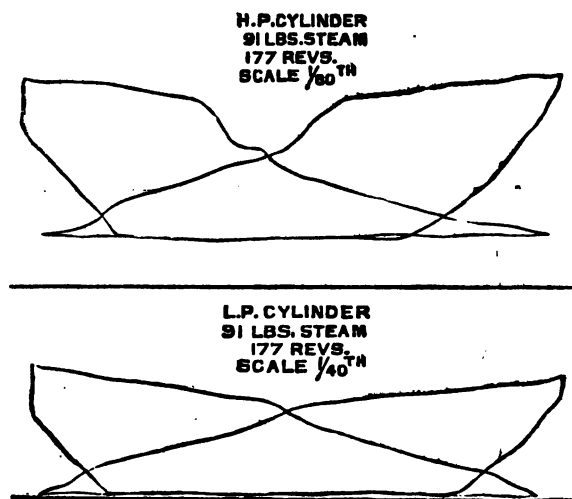
In Figs. 80 and 81 are given an excellent set of diagrams from the high and low pressure cylinders of an economical compound marine engine, the diagrams being taken with the link of the valve gear notched up or set in nearer and nearer to the position in which least motion is given to the valve, thus cutting off steam at earlier parts of the stroke. These



Figs. 80 and 81.

diagrams show a good performance, although it is not always allowed that a satisfactory variable cut-off can be obtained with the link motion. All the high-pressure diagrams give expansion curves higher than the hyperbolic curve of expansion, probably due to re-evaporation of condensed or partially condensed steam, and partly, perhaps, to large clearance spaces which represent a volume of steam not apparent in the diagram judging from the point of cut-off.

In Figs. 82 are given a set of diagrams from the high-speed Woolf compound engine, made by Messrs. J. and H. Gwynne, which drives the dynamo electric machines in the First Avenue Hotel, London. The steam cylinders are placed side by side and their piston-rods take hold of one crosshead, there being only one crank. The cylinders are 11in. and 17in. diameter respectively and the stroke 13in. The boiler pressure is 100lb., and the speed from 175 to 200 according to the number of lamps in circuit. The diagrams show very good



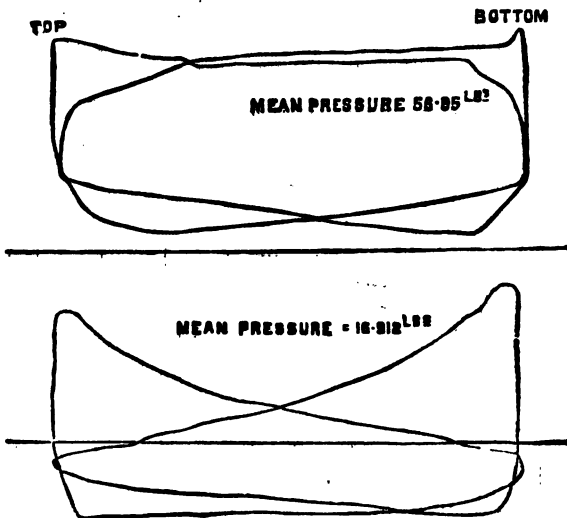
FIGS. 82.

distribution but rather too much compression even for 177 revolutions per minute. The compression line departs very much from a hyperbolic compression line, due probably to a wet condition of the steam, but it would be difficult to find a satisfactory explanation, involving as it would saturation, re-evaporation, and superheating, rapidly repeated.

The diagrams given in Figs. 83 are from the engine of the ss. "Telamon," one of the tandem compound engines of the single-crank Holt type. In these engines the high-pressure cylinder is fitted with a piston valve, and is mounted above

the low-pressure, into which it delivers the exhaust steam, controlled by a slide valve. The cylinders are 27in. and 58in. diameter, and 5ft. stroke. The diagrams were taken with the boiler pressure at 74lb., vacuum 26in., revolutions 57·5. The indicated horse-power in the high-pressure cylinder was 548, and in the low-pressure 750, a total of 1298 horse-power.

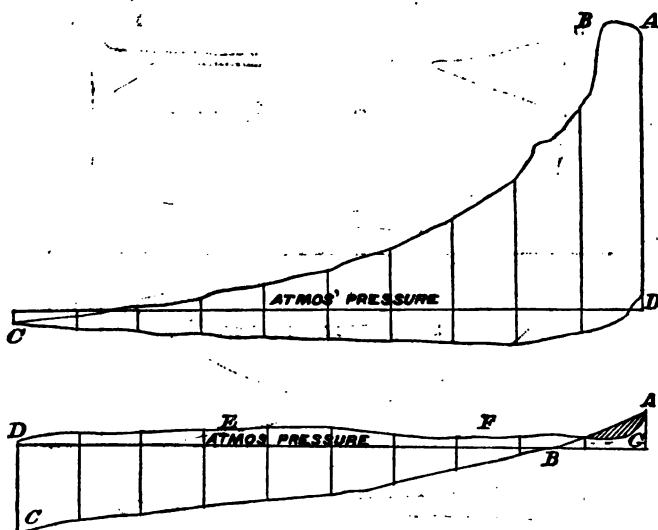
A pair of instructive diagrams is given in Figs. 84 and 85. They were taken from the engine exhibited in the Inventions



FIGS. 83.

Exhibition in 1885, by Messrs. Galloway and Sons. The engine was shown at work, and the instructions from the Exhibition executive were that the engine should indicate 185 horse-power, but it was never called upon to do more than 30 horse-power. It had a high-pressure cylinder placed in an inclined position over the low-pressure cylinder. The high-pressure diagram, Fig. 84, has the appearance of being a good diagram, and would be so if it were from a high-pressure condensing engine on light work. The engine is, however, a non-condensing engine, and

the exhaust line C D ought, of course, for efficient working of the engine, to have been considerably above instead of below the atmospheric pressure line. The work that the engine was called upon to do was, however, so very light that the automatic expansion gear, with which the engine was fitted, cut off the steam at about one-sixteenth of the stroke. The total steam thus admitted was so small in quantity that, on expansion, it fell, towards the end of the stroke, to a pressure below that of



FIGS. 84 and 85.

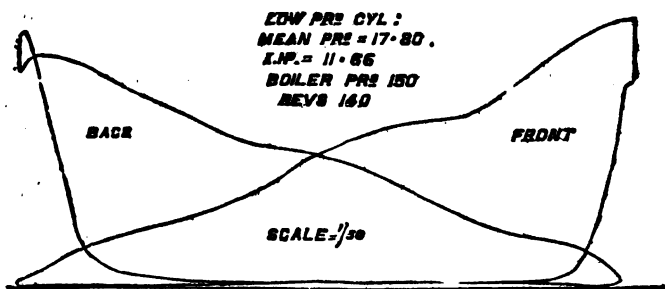
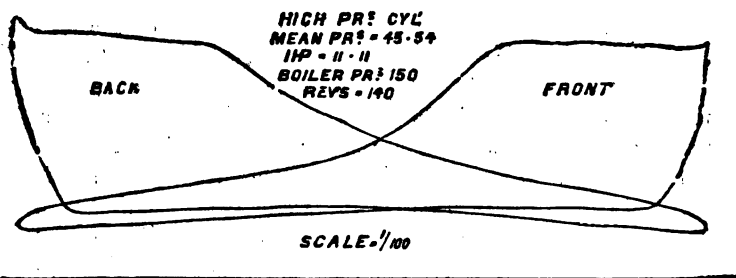
the atmosphere. At this low pressure it exhausted into the low-pressure cylinder, which started with the pressure shown at A, Fig. 85. The quantity of steam being so small for the large cylinder, its pressure fell at B about one-seventh of the stroke to the atmospheric pressure, and continued to fall through the whole stroke. At C the exhaust port opened and, instead of steam passing out, air rushed in to fill the partial vacuum, and raised the pressure to D. The air so admitted became slightly heated, and the exhaust pressure line of the low-

pressure cylinder was thus raised to D, E, F, G; a small amount of compression occurring from G. Thus the whole of the low-pressure diagram, with the exception of the very small shaded piece at G, represents negative work, and has to be deducted from the high-pressure cylinder diagram. The high-pressure diagram shows 47.6 horse-power, and the low-pressure 20.6 horse-power, the net indicated horse-power being 27.0. It will be observed that the effect of the operation of the low-pressure cylinder on that of the high-pressure was that of an inefficient condenser, in which the vacuum was produced by the air pump alone, the low-pressure cylinder and pump performing this office, worked by the high-pressure piston.

Such cases as this are not likely to occur very often, but it serves to show in a marked degree the evil effects of using a compound engine much too powerful for the work it is called upon to do. A compound engine should not be called upon to give off much more or much less than that for which its cylinders are designed.

In concluding this part of our subject we must give diagrams from the compound portable engine which with others was tested at Newcastle, by the Royal Agricultural Society in July, 1887, as referred to at page 135, and made by Messrs. Davey, Paxman and Co., to whom the Society's prize was awarded. These diagrams, Figs. 85A, are from an engine which has shown the highest efficiency yet attained by any compound portable engine. The engine may be said to be of the intermediate receiver type, with cranks 90 degrees apart. This will have been gathered from the rise in pressure of the steam on the exhaust-stroke in the high-pressure cylinders. Of the latter diagrams there is a little to be said. Of the low-pressure diagrams, it may be noted that the compression is very considerable at both ends; that at the back end the pressure produced by compression exceeds that of the incoming steam, while that at the front end is not quite so much, the sudden cessation of rise in pressure indicating, it is supposed, a critical point at which excessive moisture of the steam compressed prevents for a time the rise in temperature

and pressure. It does not, however, occur at the back end. The high-pressure cylinder of this engine was fitted with Paxman's automatic cut-off valve gear, and the low-pressure with a Trick valve. The consumption of steam by this engine was about 18·75lb. per indicated horse-power per

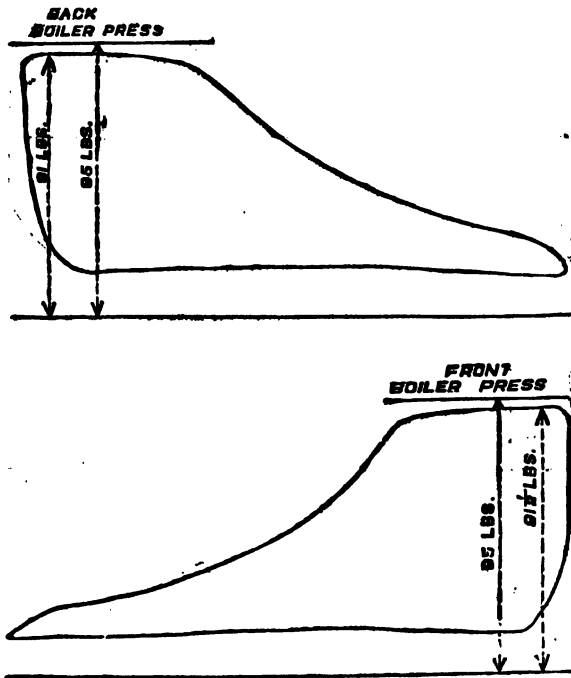


Figs. 85A.

hour, the coal consumed 1·65 per indicated horse-power, 11·38lb. of water being supplied to the boiler per 1lb. of coal consumed. The coal was best Welsh, and the stoking was, of course, of the most perfect kind. (See pp. 125 and 134.)

For comparison with this remarkable performance of a small non-condensing engine, and with that of larger condensing

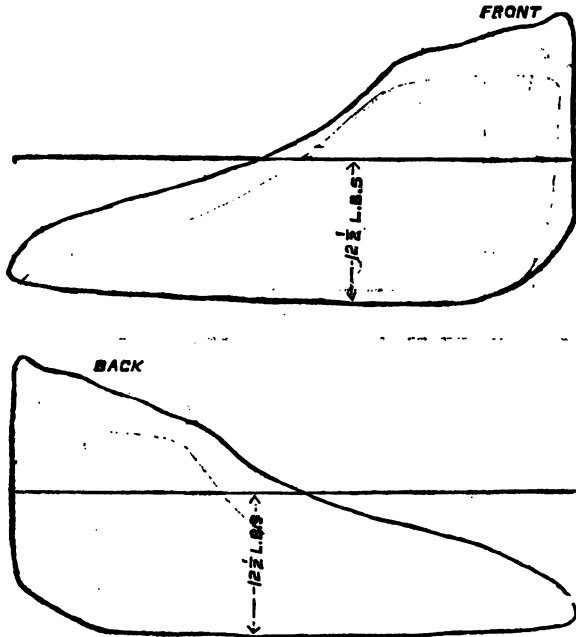
compound engines in ordinary work, attention may be directed to the set of diagrams Figs. 85B and 85C. These are from an engine of 2,000 indicated horse-power in the mill of Messrs. A. and A. Crompton, at Shaw, near Oldham. It is an engine of very large proportions, built by Messrs. J. and E. Wood, of



FIGS. 85B.

Bolton. The distribution of the steam to both cylinders is by means of a Corlias valve-gear, and is, as will be seen from the diagrams, very perfect. Steam is cut off, it will be noticed, at about one-third stroke in the high-pressure cylinder, and at about the same point in the low-pressure cylinders. The range of expansion is thus not great, and this, taken with the excel-

hence of the valve-gear and the general construction, probably explains the very great economy obtained. Even when working at only about 1,670 indicated horse-power, this engine consumed but 15.38lb. of steam per indicated horse-power per



FIGS. 85c.

hour. The boilers are evidently not as high in efficiency as the engines, as they only evaporated about 9lb. of water per 1lb. of coal, but even so the coal consumption is only 1.697lb. of coal per indicated horse-power per hour, or 1.896lb. of slack, costing 6s. 4d. per ton in the stokehole.

PART X.

DIAGRAMS FROM TRIPLE COMPOUND ENGINES.

For large powers, at least, the triple compound system of engine seems to be destined to displace the compound, just as the compound has displaced the simple engine or engine in which the expansion is performed in one cylinder; and we may expect that, as the means of making boilers capable of standing higher pressures are acquired, the triple expansion engine, or the engine in which the expansion is performed in three stages and three cylinders, will give way to four-stage expansion and four cylinders. There is, however, not so much possible gain from the use of the higher pressures and the quadruple compound engine as compared with the somewhat lower pressure triple compound engine, as there was by the latter instead of the ordinary compound. At least this is the commonly received belief, which is guided by reference to temperatures alone, but possibly it may be found that the higher pressures which by a given range of expansion are accompanied by a small fall in temperature may be very advantageous.

As examples of the diagrams from what are commonly called triple expansion engines, Figs. 86, 87, 88, and 89 are given. These form a set from the steam yacht "Isa," which is fitted with engines with cylinders 10in., 16in. and 28in. diameter and a stroke of 2ft. Of these diagrams Fig. 86 is from the high-pressure cylinder, and shows a maximum

pressure of 122lbs. above the atmosphere, the revolutions being 108 per minute. Fig. 87 is from the intermediate cylinder. It does not, however, exactly correspond with that from the high-pressure cylinder, as it was taken when the steam pressure had fallen to 93lbs. and the revolutions to 104. The diagram Fig. 88 was taken from the low-pressure

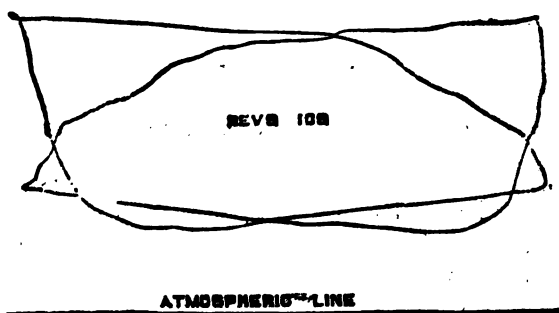


FIG. 86.

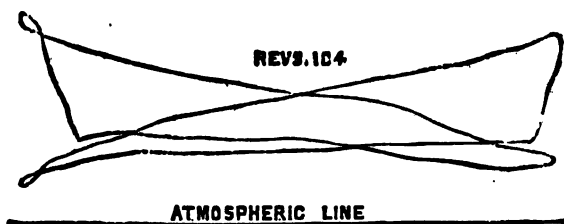


FIG. 87.

cylinder when the boiler pressure was 85lbs. and the revolutions 100, and Fig. 89 was taken with pressure at 110 and revolutions 102. At these various pressures and speeds the powers indicated were: Fig. 86, high pressure 64.2 horse-power; Fig. 87, intermediate 23.8 horse-power, and Fig. 89, low pressure 98 horse-power; or a total gross indicated horse-

power of 186, with the normal pressure of 120lbs. The engine indicates about 200 horse-power and consumes about 300lbs. of Welsh coal per hour. It should be mentioned that the high-pressure cylinder is mounted above the intermediate cylinder, the two pistons being arranged in tandem and working on one crank, the lower pressure piston working on the second crank. The total pressures on the cranks are thus approximately the same. The diagrams from the high-

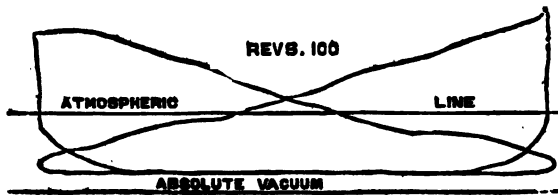


FIG. 88.

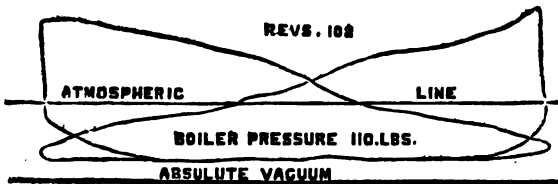


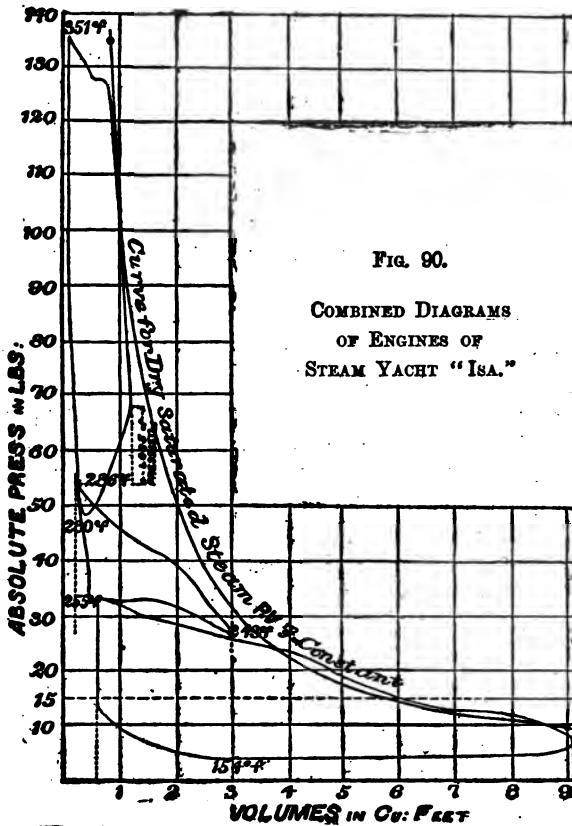
FIG. 89.

pressure cylinder have, it will be seen, the slowly falling back pressure line due to the gradual lowering of pressure as the steam passes to the intermediate cylinder. In the left hand diagram compression commences earlier than in the right hand diagram, the difference being partly intentional, the object being to provide a more powerful cushion below the piston than above it, because gravity acts on the down stroke to increase the work to be done in bringing the reciprocating parts to rest, whilst on the up stroke it similarly decreases

the necessity for cushioning at the upper end of the cylinders. It is, however, more than is desirable or economical in the left hand diagram (Fig. 86), as shown by the loop, which indicates that steam was compressed to a pressure greater than that of the incoming steam. Work was thus done upon it from which an equivalent in work was not obtained. The slide-valve was too low, and hence closed the exhaust-port earlier at that end of the cylinder than was desirable. A similar defect is observable in the diagram from the upper end of the intermediate cylinder. The power given off by the intermediate cylinder is comparatively small, and from the power point of view alone might be supposed to show that the intermediate cylinder could be usefully replaced by slight addition in the size of the other cylinders. It must not, however, be forgotten that this cylinder plays its part in reducing the range of temperature in any cylinder, and it is chiefly due to this reduction that high pressure may be more efficiently used in triple and quadruple cylinder compound engines. It will also be observed that the high and intermediate pressure cylinders together do very nearly the same amount of work upon the crank to which their pistons are connected, as is done by the low-pressure piston upon the other crank. The engines of the "Isa" were amongst the first triple engines ever put into a vessel. They were designed by Mr. Taylor, of Sunderland, and their performance has been exceedingly satisfactory. The "Isa" is the property of E. C. Healey, Esq.

Fig. 90 is an instructive combined diagram, showing at a glance the performance of the steam in the three cylinders of the "Isa" engines. This diagram was made from the originals of Figs. 86 to 89 by Mr. J. Welsh, a student under Professor Jamieson, and is a very good example of diagrams reduced to one scale. The theoretical curve of expansion is the adiabatic curve of dry saturated steam, calculated according to the expression $PV^{\frac{10}{9}} = \text{a constant}$, or, in other words, on the assumption that the pressure raised to the 9th power

varies inversely as the volume raised to the 10th power. The curve is the same as that described at pp. 85—86, in which the vertical ordinates have heights in proportion, as there



given, to the expression $p' = \frac{P}{R^{10}}$. The clearance spaces are assumed to be $\frac{1}{10}$, $\frac{1}{14}$, and $\frac{1}{18}$ respectively of the capacity of

the high, intermediate and low-pressure cylinders. The chief figures relating to these engines and diagrams may be thus tabulated :—

STEAM YACHT "Isa," TRIPLE EXPANSION.

| | CYLINDERS. | | |
|-----------------------------|----------------|------------------------|---------------|
| | High Pressure. | Intermediate Pressure. | Low Pressure. |
| Diameter of Cylinders | 10in. | 16in. | 28in. |
| Stroke..... | 2ft. | 2ft. | 2ft. |
| Volume of Cylinder | 1·09c. ft. | 2·79c. ft. | 8·55c. ft. |
| Volume of Clearance | 0·1c. ft. | 0·2c. ft. | 0·54c. ft. |
| Range of Temperatures | 71 F. | 43 F. | 101 F. |
| Indicated Horse-power | 64·2 | 23·8 | 98·0 |

Some very interesting examples of indicator diagrams from engines of an ingenious and peculiar arrangement are afforded by those which follow, Figs. 92 and 93. To explain them, however, it is necessary to give a diagram section of the cylinders, Fig. 91, as the engine might otherwise, from the diagrams, be supposed to be a triple compound engine, and not a single-acting compound. The engine is made by Messrs. Willans and Robinson, and it has been described by Mr. Willans as follows :—

"In Fig. 91, H is the high-pressure cylinder, R is the receiver, and L is the low-pressure cylinder. Fig. 92 shows the diagrams from these three spaces, the arrows show the direction in which the piston is travelling as each line is traced. The steam enters—and is expanded in—H during the downstroke; in the course of the upstroke it is transferred to R; and during the early part of the following downstroke it enters L, and during the latter part of the stroke it expands there. During the next succeeding upstroke it passes away into the atmosphere or to a condenser. Each of the three variable spaces H, L, and R, must be indicated, because the varying pressures in each affect the power of the engine. If there is variation

in pressure on the lower side of the high-pressure piston it must tell for or against the engine. It cannot be ignored.

“Engineers do not usually indicate the variations in pressure in the receiver of an ordinary double-acting compound engine, for the cylinder diagrams already show them so far as they affect each side of each piston. In this case, however, unless we take a diagram from the lower side of the high-pressure piston, we have no means of ascertaining the varying pressures which affect it during the upward and downward

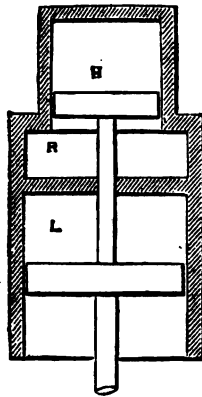


FIG. 9L.

movement. There is therefore every reason for taking such a diagram.

“Until, on the downward stroke, the steam is cut off from L, the capacity of R is increasing and the pressure there is falling in consequence. As soon as steam is cut off from L the pressure in R rises, because the spaces in connection with it decrease until the end of the downstroke. As the piston moves upward again the pressure remains approximately constant—there is a slight difference, due to the piston-rod, but we need not consider this. The result is the triangular-shaped

L

diagram shown, and the point at issue is whether it is to be added to or subtracted from the power. Let us turn for a moment to the low-pressure cylinder; the atmospheric pressure on the under side of the low-pressure piston acts for the engine on the upstroke and against the engine on the downstroke, and as it is equal in each case it is not usually indicated. But if on the downstroke only the atmospheric pressure could be wholly or partially removed, there would be a gain in power, because there would be less pressure on the under side of the piston against the engine than there was for

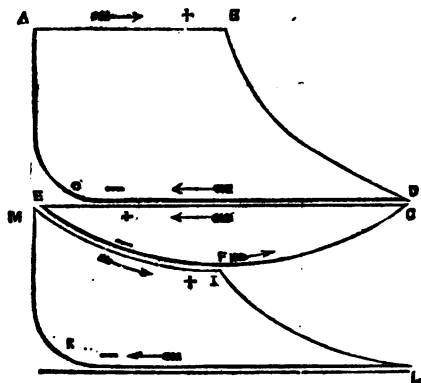


FIG. 92.

it. In the case of the high-pressure cylinder, in Fig. 91, these are the precise conditions: During the downstroke the pressure acting against the engine, and indicated by the lower line of the receiver diagram E F G, is less than that acting for the engine on the upstroke, and shown by the upper line E G. The actual pressure in the receiver is not added to the power of the engine, but the power which is developed by the variations in pressure under the high-pressure piston is taken into account in the same manner as is the power developed in a similar way above it.

“Treating the matter as a ledger account, we have :—

| | In space. | For the engine. | Against the engine. |
|---|-----------|---|--|
| 1 | H { | Pressure indicated by line A B D (Fig. 92). | { Pressure indicated by line A C D. Balance shown on H.P. diagram. |
| 2 | L { | Pressure indicated by line M I L. | { Pressure indicated by line M K L. Balance shown on L.P. diagram. |
| 3 | R { | Pressure indicated by line E G. | { Pressure indicated by line E F G. Balance shown on R. diagram. |

The three diagrams thus show the balance of power after deducting back pressure.

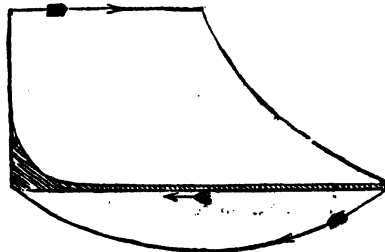


FIG. 93.

“The high-pressure cylinder diagram shows the mean pressure in that cylinder, but in explaining the action of the steam from the point of view of its positive and negative action on the piston at any one time, what is beneath it must be considered. If you take the pressures above and below the piston on the downstroke in the same way as you take them on the upstroke, you will find that they show a power considerably in excess of that shown by the high-pressure diagram. Fig. 93

shows this graphically. The outline of the whole diagram gives the true net pressures for the engine on the downstroke. The outline of the sectioned central part shows the true net pressures against the engine on the upstroke."

The engine is thus a compound engine with single-acting pistons, but as the negative pressure on the lower side of the high-pressure piston varies, there is virtually a triple stage expansion of the steam, viz., 1. During part of the descent of the high-pressure piston, and taking place in H. 2. Slight expansion in passing from H to R as the piston rises, during which time it is doing no work, the high-pressure piston being in equilibrium. 3. Expansion in passing from R to L, performing work on low-pressure piston, and causing the descent in pressure in R, which counts in favour of H.



PART XI.

TRIPLE STAGE AND QUADRUPLE-STAGE EXPANSION OF STEAM.

It is unnecessary to multiply examples of diagrams from triple compound engines, as they differ only in detail from those we have given, and we are not dealing with the comparative efficiency of engines of different designs or of different detail. As, however, the extension of the employment of very high pressures is more marked than at any previous period, it may be here of some service to point out what may be expected to be gained by using the higher pressures, and, therefore, the triple and quadruple compound engines. We say therefore, because high ranges of expansion are not economical in one cylinder, because great variation of temperature in a cylinder is productive of wasteful condensation. From a thermo-dynamic point of view, the greatest possible efficiency may be taken as proportional to $\frac{T-t}{T}$. Applying

this, we may take for example engines working under several different pressures, and an assumed condenser or lowest pressure of 7lbs. Further on we shall see the effect of taking a condenser pressure of 1lb. and temperature of 100deg. F.

THEORETIC EFFICIENCY OF SIMPLE, COMPOUND, TRIPLE, AND QUADRUPLE ENGINES.

| | Simple. | Compound. | Triple. | Quadruple. |
|------------------------------------|------------|------------|-------------|-------------|
| Boiler pressure | 20lbs. ... | 75lbs. ... | 150lbs. ... | 200lbs. ... |
| Total pressure..... | 35lbs. ... | 90lbs. ... | 165lbs. ... | 215lbs. ... |
| Condenser pressure..... | 7lbs. ... | 7lbs. ... | 7lbs. ... | 7lbs. ... |
| Temperature of steam = T | 260 ... | 320 ... | 366 ... | 388 ... |
| Temperature of condenser = t | 177 ... | 177 ... | 177 ... | 177 ... |
| Range of temperature ... | 83 ... | 143 ... | 189 ... | 211 ... |
| Efficiency = E | 0.115 ... | 0.183 ... | 0.228 ... | 0.248 ... |
| Efficiency ratio | 1 ... | 1.6 ... | 2.0 ... | 2.156 ... |
| Gain, per cent. | 0 ... | 30 ... | 50 ... | 53 ... |

There is thus from this point of view a possible heat engine gain in favour of the compound condensing engine with 90lbs. steam, as compared with the simple engine with 35lbs. steam, of 30 per cent., and of $50 - 30 = 20$ per cent. in favour of the triple, as compared with the compound, and of $53 - 30 = 23$ per cent. as the heat efficiency of the quadruple engine, as compared with the two-stage compound engine, but of only $53 - 50$ in comparison with the triple compound—that is, with the pressures assumed. It is not likely that 215lbs. will be very soon exceeded, but the quadruple engine may for large powers become common, because the practical gain as a result of performing the expansion by several comparatively small ranges of temperature is very considerable, a fact which does not enter into the calculations above as to thermal efficiency, for by this comparison the efficiency of 215lbs. steam would be apparently the same if used and expanded down to 7lbs. in one cylinder as if expanded in three or four cylinders. It would seem, however, to be obvious that above 200lbs. the possible gain from increased pressures becomes less and less, because the rate at which the pressures increase is more rapid as the higher temperatures are reached than at the lower temperatures, so that the work which can be performed by a lb. of steam, or at all events the efficiency of a lb. of steam, does not increase so rapidly above 200lbs. as it does from low pressures up to 200lbs. In the practical engine, however, the fact that steam may be expanded down from 215lbs. pressure to 165lbs. with a reduction of only 22deg. in temperature may prove of importance.

The hypothetic thermo-dynamic efficiencies, with steam at various pressures and temperatures, are better shown by Fig. 94, which shows the ideal efficiency of steam up to 300lbs. per square inch, and is taken from a paper in the *Transactions* of the Institute of Naval Architects, 1886, by Mr. W. Parker, M.Inst.C.E. The condenser temperature assumed for the purposes of this diagram is 100 degrees. It will be seen that, with steam at 50lbs., the efficiency is 0.261, and that it rises gradually as the pressure increases, so that at 60lbs. it is 0.270 ;

at 100lbs. it is 0.298 ; at 150lbs. 0.321 ; at 200lbs. 0.339 ; at 250lbs. 0.353 ; at 300lbs. 0.365. From these figures it will be observed that Mr. Parker's calculations lead to the expectation that "in advancing from 60lbs. to 150lbs. the efficiency ought to be increased by 19 per cent., or, what is the same thing, to effect an economy of 16 per cent. The fact that an increase of economy is actually obtained considerably above this amount when using the triple expansion engine shows that this engine is working under conditions more nearly approaching to those required for the maximum efficiency than the other," i.e., the

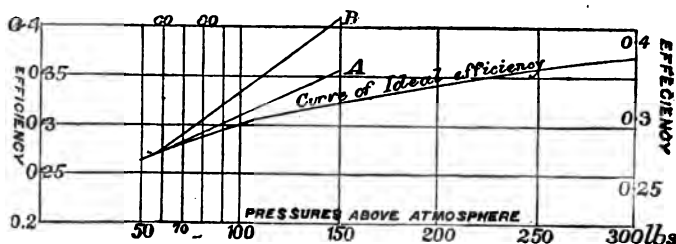


FIG. 94.

lower pressure engine with which the comparison is made. "The line to A in the diagram lying above the ideal efficiency curve represents the height to which the efficiency has gone, starting from 60lbs. if 25 per cent. economy has been effected; while the line to B shows the similar height upon the assumption of an economy of 33 per cent."

The reader who wishes to follow up this subject and the cognate questions concerning condensation in the cylinders of compound and triple compound engines may be referred to the *Transactions* above mentioned.

Special reference may also be made to a most important paper on this subject read at the Institution of Civil Engineers, on the 13th March, 1888, by Mr. P. W. Willans, M.Inst.C.E. A very extensive series of economy trials were carried out with one of Mr. Willans' central-valve triple expansion non-condensing vertical engines, having one crank

and three cylinders in line. By moving one or both of the upper pistons the engine could be easily changed into a compound or into a simple engine at will. Distinct groups of trials were thus carried out under favourable circumstances for a satisfactory comparison of simple, compound and triple working, with various steam pressures and at various speeds.

The trials were specially arranged so as to obtain information concerning the steam condensed in each cylinder, and the steam re-evaporated therein, by comparison with the quantity of steam represented by the volume of the cylinder or cylinders at the point of cut-off. From these figures were obtained the difference between the quantity of steam used in doing the indicated work and the quantity apparently necessary for it, calculated by a formula deduced from Mr. MacFarlane Gray's investigations on steam. The difference or missing quantity is thus found. The formula is—

$$U = \left(\frac{1438 - 0.7 T}{T} + \frac{T - t}{T + t} \right) T - t.$$

That is to say, working with steam at absolute temperature T , and expanding to absolute temperature t , and exhausting against the pressure due to steam at the latter temperature, the heat units U thermo-dynamically due in the shape of work from 1 lb. of steam are as given in the above expression. Then, taking the heat unit as = 772 foot-pounds, 1 horse-power for one

$$\text{hour} = \frac{33000 \times 60}{772} = 2564 \text{ heat units.}$$

The weight w of steam in lbs. per horse-power per hour theoretically required is therefore—

$$w = \frac{2564}{U}.$$

Mr. Willans' experiments proved the great advantage of high pressures, and of small ranges of expansion in one cylinder, and therefore the necessity for compound, triple, or quadruple expansion engines.

PART XII.

DIAGRAMS FROM LOCOMOTIVE ENGINES.

THE behaviour of steam in the cylinder of a locomotive affords at least as much, and probably much more, useful and suggestive information than in any other engine, for in it not

LINE OF STEAM CHEST PRESSURE.

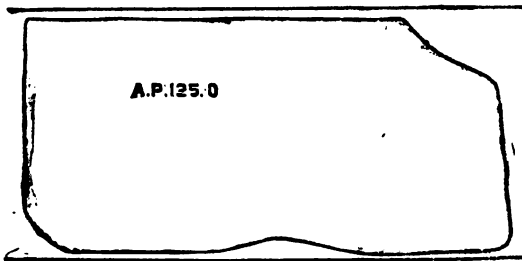


FIG. 95.

Speed, 8 miles per hour ; Cut-off, 79 per cent. ; Steam chest pressure, 140lbs. ; Boiler pressure, 140lbs. ; Gradient, 1 in 264 up ; Load, 73 ; Approximate weight of engine and train, 795 tons ; Revolutions per minute, 23.40 ; Total I.H.P., 240.75.

only is the pressure employed higher than in any other simple engine, but greater range of variation in the cut-off occurs. As illustrative of locomotive performance and affording excellent examples of diagrams from locomotives of very high efficiency, the diagrams Figs. 95 to 105 are here given. They are from a goods engine of the powerful class C type of the

London Brighton and South Coast Railway, designed by Mr. W. Stroudley, M.Inst.C.E. The cylinders are 18.25 diam., 26in. stroke, six wheels coupled, and 5ft. in diameter; the boiler pressure is 140lbs. The average pressure throughout the stroke is given on each diagram, and under each are given the particulars concerning them, as published in *The Engineer*, Vol. LX., p. 390. They are all from the front end of the left hand cylinder.

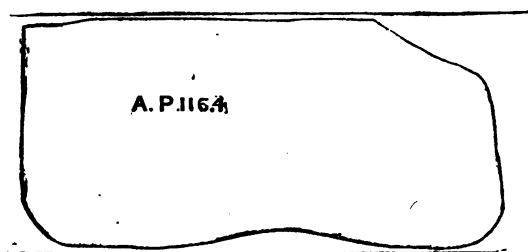


FIG. 96.

Speed, 8 miles per hour; Cut-off, $75\frac{1}{2}$ per cent.; Steam chest pressure, 134lb.; Boiler pressure, 135lbs.; Gradient, 1 in 264 up; Load, 73; Approximate weight of engine and train, 795 tons; Revolutions per minute, 45.44; Total I.H.P., 358.74.

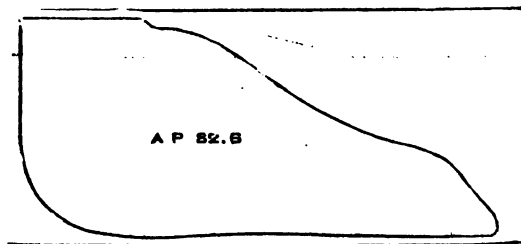


FIG. 97.

Speed, 10 miles per hour; Cut-off, 36 per cent.; Steam chest pressure, 135lbs.; Boiler pressure, 135lbs.; Gradient, 1 in 264 up; Load, 73; Approximate weight of engine and train, 795 tons; Revolutions per minute, 56.80; Total I.H.P., 318.16.

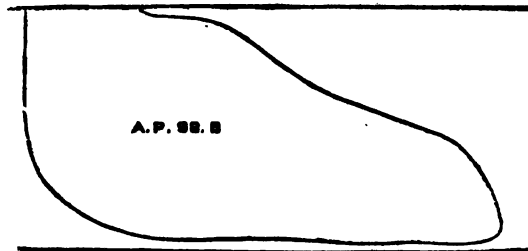


FIG. 98.

Speed, 12 miles per hour; Cut-off, 36 per cent; Steam chest pressure, 140lbs.; Boiler pressure, 140lbs.; Gradient, 1 in 264 up; Load, 73; Approximate weight of engine and train, 795 tons; Revolutions per minute, 68.16; Total I.H.P., 456.84.

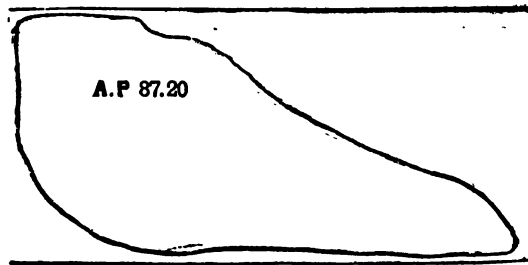


FIG. 99.

Speed, 13 miles per hour; Cut-off, 26 per cent.; Steam chest pressure, 140lbs.; Boiler pressure, 140lbs.; Gradient, 1 in 264 up; Load, 73; Approximate weight of engine and train, 795 tons; Revolutions per minute, 73.84; Total I.H.P., 436.69.

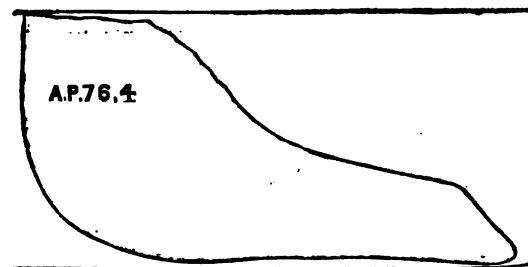


FIG. 100.

Speed, 10 miles per hour; Cut-off, 23 per cent.; Steam chest pressure, 140lbs.; Boiler pressure, 140lbs.; Gradient, 1 in 264 up; Load, 73; Approximate weight of engine and train, 795 tons; Revolutions per minute, 56.80; Total I.H.P., 294.28.

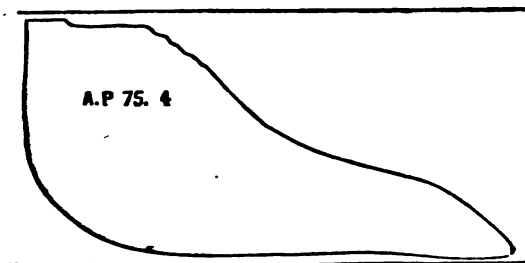


FIG. 101.

Speed, 13 miles per hour ; Cut-off, $23\frac{1}{2}$ per cent. ; Steam chest pressure, 139lbs. ; Boiler pressure, 140lbs. ; Gradient, 1 in 264 up ; Load, 73 ; Approximate weight of engine and train, 795 tons ; Revolutions per minute, 73.84 ; Total I.H.P., 377.60.

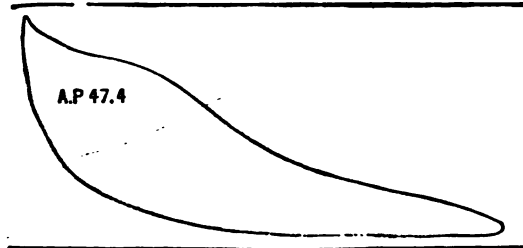


FIG. 102.

Speed, 27 miles per hour ; Cut-off, 17 per cent. ; Steam chest pressure, 140lbs. ; Boiler pressure, 140lbs. ; Gradient, 1 in 264 down ; Load, 73 ; Approximate weight of engine and train, 795 tons ; Revolutions per minute, 153.36 ; Total I.H.P., 493.14.

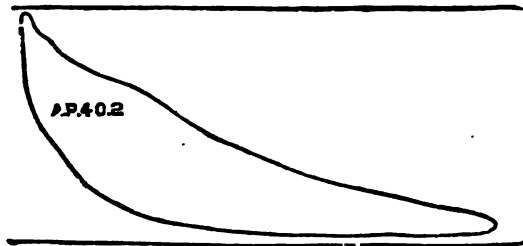


FIG. 103.

Speed, 33 miles per hour ; Cut-off, $23\frac{1}{2}$ per cent. ; Steam chest pressure, 135lbs. ; Boiler pressure, 135lbs. ; Gradient, 1 in 264 down ; Load, 73 ; Approximate weight of engine and train, 795 tons ; Revolutions per minute, 187.44 ; Total I.H.P., 511.34.

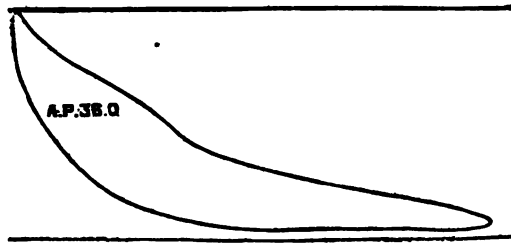


FIG. 104.

Speed, 37 miles per hour ; Cut-off, $23\frac{1}{2}$ per cent. ; Steam chest pressure, 130lbs. ; Boiler pressure, 130lbs. ; Gradient, 1 in 264 down ; Load, 73 ; Approximate weight of engine and train, 795 tons ; Revolutions per minute, 210.16 ; Total I.H.P., 513.50.

As illustrations of diagrams from a compound express passenger engine, Figs. 105 and 106, from one of Worsdell's

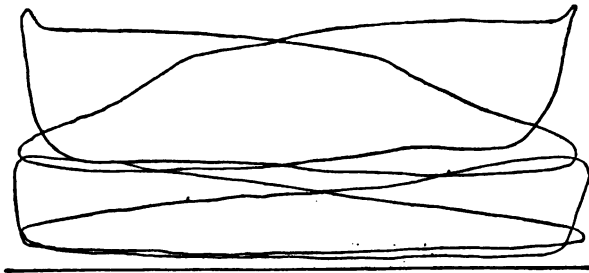


FIG. 105.

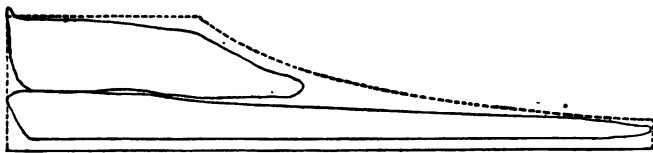


FIG. 106.

patent engines on the Great Eastern Railway, are given. Fig. 105 contains the high and low pressure diagrams superposed. The engine has a high-pressure cylinder, 18in.

diameter, and low-pressure 26in. diameter, both of 2ft. stroke. The driving wheels are 7ft. diameter, coupled to a pair of trailing wheels of the same diameter. The diagrams, as at Fig. 105, were taken when the engine was making 42 miles per hour with twelve coaches, and cutting off steam in the high-pressure cylinder at 62 per cent. of the stroke. The mean effective pressures were:—High pressure, back end, 43·4; front end, 44·1; low pressure, front, 22·6; back, 26·3; indicated horse-power, high pressure, 256·51; low pressure, 264·33; total, 520·84 indicated horse-power. Fig. 106 shows this same pair of diagrams drawn to the same scale, and showing them in comparison with the theoretical curve of adiabatic expansion. The latter is, however, plotted without allowance for the clearance spaces, and is in so far unsatisfactory; but it gives a sufficient general indication of the engine performance.



PART XIII.

GAS ENGINE DIAGRAMS.

WE have now to deal with the diagrams from gas engines of several kinds, as they present an important part of our subject. In order that the diagrams from these engines may be understood, it is necessary that some knowledge of the construction and operation of the engines should be first acquired. This may be briefly given, and the better by dividing the numerous forms of engines now in use into their representative types. These we may divide into three classes, which will be to a great extent chronological as well as descriptive. Following Mr. Dugald Clark in his description of the three main types, we have—

(1) An engine drawing into its cylinder gas and air at atmospheric pressure for a portion of its stroke, cutting off communication with the outer atmosphere, and immediately igniting the mixture, the piston being pushed forward by the pressure of the ignited gases during the remainder of its stroke. The instroke then discharges the products of combustion.

(2) An engine in which a mixture of gas and air is drawn into a pump, and is discharged by the return stroke into a reservoir in a state of compression. From the reservoir the mixture enters into a cylinder, being ignited as it enters without rise in pressure, but simply increased in volume, and following the piston as it moves forward; the return stroke discharges the products of combustion

(3) An engine in which a mixture of gas and air is compressed or introduced under compression into a cylinder or space at the end of a cylinder, and then ignited while the volume remains constant and the pressure rises. Under this pressure the piston moves forward, and the return stroke discharges the exhaust. Types 1 and 3 are explosion engines, the volume of the mixture remaining constant while the



FIG. 107.

pressure increases. Type 2 is a gradual combustion engine, in which the pressure is constant, but the volume increases.

Type 1 is represented by the Lenoir gas engine, and by several now made, mostly of small sizes, and the pistons of which draw the combustible charge in during a part of the outstroke; the charge being then ignited, the piston is caused to complete its stroke. The ideal indicator diagram from such an engine would be as shown in Fig. 107, and the actual

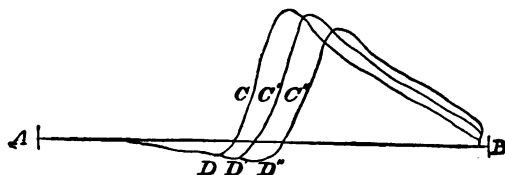


FIG. 108.

indicator diagram, from a Lenoir engine with an 8·375in. cylinder and 16·25in. stroke, is shown at Fig. 108. The stroke is represented in the diagram by the line A—B. During the part of the outstroke from A to D, D', or D'', the piston, in drawing in the mixed charge of gas and air, moved faster than the charge under the pressure of the atmosphere could follow it, and hence a partial vacuum was formed, as shown by the admission line descending to D, D', or D'' below the

atmospheric line. At these points ignition took place, and the pressures rose as shown by the lines C, C', C". Fig. 108 contains the diagrams of three strokes.

Of the second type of engine few are being made, one of them being that of Mr. Simon, of Nottingham, in which the mixed charge is ignited in the compression chamber at the commencement of the stroke of the power piston. Fig. 109 is

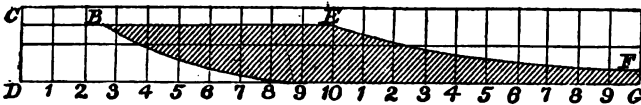


FIG. 109.

a hypothetic diagram showing the action of this engine. For convenience the pump diagram is shown with the power diagram, and 9, B, C, D is the pump diagram. Air and gas are drawn in as shown by line D A. On the return stroke of the pump piston, the mixture is compressed from 9 to B, and then being equal to the reservoir pressure, it is forced into the reservoir. G, B, E, F is the motor diagram. From C to E the mixture is flowing from the reservoir and following up the motor piston,

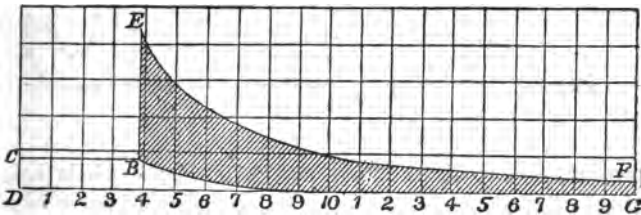


FIG. 110.

combustion taking place the whole time. At E communication with the reservoir is cut off, the remainder of the stroke being performed by the expanding and cooling gases which are ejected on the return stroke of the motor piston.

Of the third type of engine many are now made, including the Otto, the Clerk, the Atkinson, the Beck and the Griffin, besides several others in France and Germany. Fig. 110 is a hypothetical diagram from an engine of this type.

In this, as in the Figs. 107 and 109, the line D G represents the whole stroke of the motor piston. Suppose an engine of this type to draw into a pump a certain quantity of air, or air and gas, represented in Fig. 110 by D, 10, the pump diagram for convenience being placed in the motor diagram as in Fig. 109. On the return stroke the air is compressed from 10 to B, and from B to C is delivered into a reservoir at about 40lbs. per square inch; 10, B, C, D is thus the pump diagram. The motor piston moves from C to B, followed up by the air and gas from the reservoir. At B the charge is ignited and the

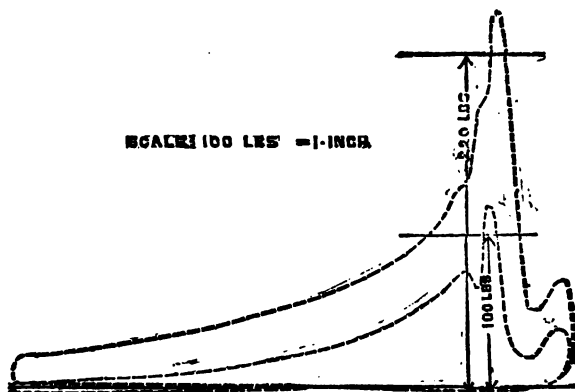


FIG. 111.

pressure rises to about 220lbs. and 2,790 E, the whole height of the diagram representing 250lbs. From this point expansion of the exploded charge continues to the point F, where exhaust takes place.

Of this type the horizontal engine of the British Gas Engine Company is an example, and Fig. 111 is a diagram from this engine working with a half charge and full charge, the engine being regulated by a governor and igniting arrangement which vary the quantity of the charge instead of the strength, or instead of cutting off the gas altogether, as in some engines. The admission part of the stroke is seen in both the indicator

diagrams on Fig. 111, the fall in pressure from that of the reservoir being due to the expansion from the small reservoir into the cylinder until the point of cut-off and of ignition. These diagrams show exceedingly satisfactory combustion and expansion, especially with the half charge.

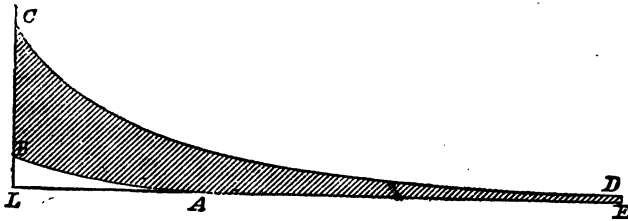


FIG. 112.

Fig. 112 illustrates hypothetically the action of the most perfect form of this third type. Instead of a separate reservoir, a space is left at the back end of the cylinder, into which the piston does not enter, and into this space the combustible

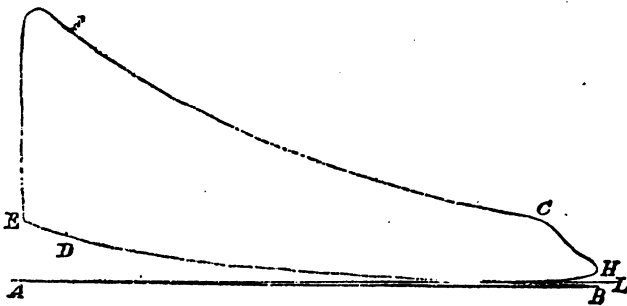


FIG. 113.

mixture is compressed by the power piston itself. The result is that ignition can be effected at the commencement of the power stroke, the whole of which is an effective working stroke. The most extensively used of this type is the Otto engine. Fig. 113 is a normal diagram from a 10 horse-power engine.

The line A, L, is the atmospheric pressure line. The indications of the diagram are best understood by following the cycle of operations of the engine. Beginning at the point A, the lowest line of the diagram to B, represents the pressure during the first forward stroke while gas and air are entering the cylinder; this line lies at a nearly uniform distance below the atmospheric line. This line A, B, shows a negative pressure at 0.15 atmosphere. At B, the inlet valve closes, and by the return stroke the gases are compressed into the compression space at the back end of the cylinder. This compression is represented by the line B, G, D, E, which crosses the atmospheric line at G, and shows a pressure of two atmospheres at E. One revolution of

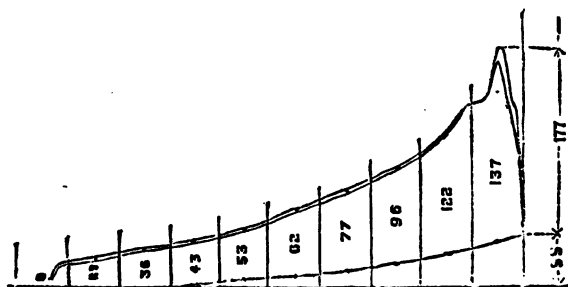


FIG. 114.

the engine is now completed, and the charge is ignited just as the crank is passing the centre. The rapid burning of the gas generates a large amount of heat, increasing the temperature and pressure, which reaches about 135lbs. as a maximum. The gases now expand during the second forward stroke, and exert upon the piston energy, which, by means of a fly-wheel, carries the engine through the remainder of the cycle. At C, the exhaust-valve opens, allowing the burnt gases to escape. The line H, G, A, shows the pressure while these gases are being expelled by the second return stroke of the piston. Only one working stroke is thus obtained in two revolutions.

Fig. 114 is from one of Clerk's gas engines of 12 horse-power. In this engine the whole is completed in one revolu-

tion, and an impulse given at every outward stroke of the power piston. The engine contains two cylinders, the one, called the displacer cylinder, taking in the combustible charge and transferring it to the power cylinder. At the end of the power cylinder is a long conical space, which at its smaller end is in communication with the displacer cylinder with intermediate controlling valves. At the other end of the power cylinder are ports which are uncovered by the motor piston when near the outer end of its stroke, and act as exhaust ports. The pistons of both cylinders are worked by connecting rods from the same crankshaft, the displacer crank being in advance of the power crank by a right angle. The displacer piston during its outer stroke takes in its charge of gas and air. It then makes its return stroke, and has performed a short portion of it by the time that the power piston has uncovered the exhaust ports, and has almost completed its in stroke by the time that the power piston has again covered the exhaust ports. The greater part of the contents of the displacer cylinder has thus been sent into the power cylinder, and has driven before it the products of the just-completed power piston stroke, and leaving the power cylinder full of a combustible charge. The power piston now returns and compresses the charge into the conical space already referred to, ignition takes place, and the out stroke is again performed by the power piston. Every out stroke of the power piston is thus a working stroke. The 12 horse-power engine from which the diagram was taken has a cylinder 9in. diameter, 20in. stroke, makes about 132 revolutions per minute; mean pressure, 66·1lbs.; maximum pressure, 232lbs.; pressure before ignition, 55lbs.; indicated horse-power, 28·01; brake horse-power, 23·21; consumption of gas per indicated horse-power, 20 cubic feet; consumption per brake horse-power, 24·12 cubic feet; total consumption per hour, running light, 90 cubic feet. It will have been seen from the brief description of the working of the engine that a diagram from the displacer cylinder will not represent work, which can be directly deducted from the power cylinder by subtracting its

area from that of a diagram from the power cylinder. Nearly all the work, however, which the displacer diagram represents is work which the power piston has to perform, and must be deducted.

Fig. 115 is a displacer diagram from an 8 horse-power engine. The 8 horse-power engines indicate about 17·4 horse-power, and give a brake horse-power of 13·7, a difference between indicated and actual horse-power of 3·7 indicated horse-power. The displacer diagram, Fig. 115, shows about 1·7 horse-power, which taken from 3·7 leaves about 2 indicated horse-power as the friction of the engine. The friction is about 11 per cent. of the indicated.

There are several other gas engines whose cycles, and, therefore, whose diagrams, are similar to those that have now been described. A very remarkable and economical gas

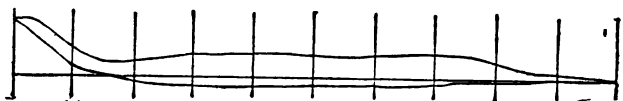


FIG. 115.

engine is designed by Mr. Atkinson and known as the differential gas engine, because the space in the cylinder for the compression and the subsequent expansion of the gases is provided by the differential movements of two pistons working in the one cylinder. The engine is of type 3, and the diagram much the same as that from a Clerk or an Otto engine.

Mr. Atkinson has recently produced a new gas engine which is a most remarkably economical engine. Its piston receives an impulse at every revolution of the crank, but the piston makes four strokes to every revolution by a motion which is communicated to it by connecting the piston-rod and the connecting-rod to a toggle joint. The piston by this means makes a short stroke drawing in air and gas and then returns compressing the charge. The charge is then ignited and the piston makes a long stroke expanding the hot gases through a greater range than has

hitherto been possible, and doing it in about one-eighth the time that it takes the crank to make one revolution. The engine thus gains in economy by the greater expansion, and by losing only about 18 per cent. of the heat generated through the cylinder walls to the water in the jacket, instead of about 50 per cent. as is usual with previous engines. Mr. Atkinson has worked at the problem in the most useful direction, namely, obtaining greater range of expansion and performance of the

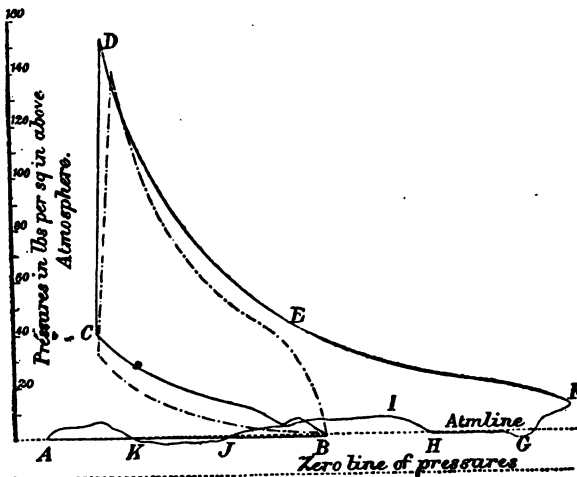


FIG. 116.

work in the cylinder as quickly as possible. He is now able to exhaust at a pressure little above that of the atmosphere, while 30lbs. to 35lbs. has been a common pressure. Fig. 116 is a diagram from this new engine.

From this we may see the work done by the piston in drawing in and compressing the charge of gas and air, and upon the piston by the explosive combustion of the charge. The first or inspiring stroke commences at A, and very closely follows the atmospheric line for some distance, but falls a little below it at B. The short out stroke there terminates, and the return

short stroke commences, and compression of the charge takes place from B to C. Ignition then occurs and the pressure rises instantly to D, as shown by the almost vertical line C D. The long out stroke is next made doing work on the piston, the expansion of the gases following the curve D E F. Exhaust then takes place with a pressure varying as shown by the line G H I J K A; the variation has, however, been removed in the latest engines. On the same Fig. 110 is a diagram in dotted lines showing to the same scale a diagram from an Otto engine using the same quantity of gas. The greater expansion obtained by the Atkinson engine gives the area E F G B in excess of that given by the Otto engine and cycle, and it is by this amount a nearer approach to the hypothetic perfection of diagram No. 112.



PART XIV.

THE MEASUREMENT OF INDICATOR DIAGRAMS.

THROUGHOUT this book only one method of measurement has been mentioned, namely, that of adding together the several pressures, usually ten, which are measured on the diagram by means of a scale or compass, and then taking the average of the arithmetic mean of these. This method is one of the simplest and perhaps most used; but an equally simple one, and one at least as accurate, is to take a piece of strip paper, and mark upon it, first the length of the first ordinate during steam admission, then place the strip upon the diagram at the place for the next measurement, putting the lower mark already made upon the upper part of the diagram, and marking upon the strip the depth of the diagram at that place. Carrying this on throughout the ten places of measurement on the diagram, the pressures at the ten places will have been added together, and they can be measured off in inches; so that if the whole measures (say) $6\frac{1}{2}$ inches, and the scale be

(say) $\frac{1}{32}$, the mean pressure will be $\frac{32 \times 6.5}{10} = 20.8\text{lb}$. When

a large number of diagrams has to be measured, this method is not only rapid and accurate, but it is one that can be so easily checked, as the number of spaces marked on the strip can be counted up and compared with the number of diagrams that have been measured.

During the past few years the use of the planimeter has very much extended, and the measurements obtained are

considered to be more accurate than the addition of a series of ordinates. This, it will be easily seen, ought to be the case, for as the planimeter measures the area of the diagram, taking every part as easily as if the diagram were a parallelogram, the areas between two ordinates of the parts which make a great angle with the horizontal will be more accurately measured than when they are taken by means of two ordinates measured by a scale. This, of course, assumes that the area measured by the planimeter is divided by the exact length of each diagram, and that the planimeter is accurately used.

It is not thought necessary to describe the planimeter here or the method of using it, for this is described in the instructions sent out by the makers with every instrument, and the theory of its action has been described in many places. They are supplied by Messrs. Elliott Bros., and a very neat form of the instrument is made by the Crosby Indicator Company, and it is specially useful in measuring the small diagrams usually taken from high-speed engines.

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